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DEVELOPMENT OF SPECIFICATIONS FOR A MOTORCYCLE DYNAMOMETER AND MOTORCYCLE COOLING SYSTEM

VOLUME I. DESIGN STUDY

OLSON LABORATORIES, INCORPORATED

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16. ABSTRACT

This project developed the specifications for a motorcycle dynamometer and motorcycle cooling system to be used in motorcycle exhaust emission certification programs. In the development of dynamometer specifications, various power absorbers, roll assemblies, and inertia assemblies were evaluated and their performance related to road data. Variable-flow blower systems were examined as a technique to simulate on-road engine cooling. Specific cooling system parameters studied included blower style, ducting requirements, noise levels, efficiency, power requirements, flow control methods, cost, and delivery.

This volume, the design study, presents the information leading to the motorcycle dynamometer specifications.

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PREFACE

This dynamometer development report is a compilation of the task report submitted to the EPA by Olson Laboratories, Inc., in partial fulfillment of Contract No. 68-03-2141.

The report is being published in the present to expedite dissemination of the information. However, since this form does not incorporate current EPA opinion or possible recent modifications, the opinions, conclusions, and recommendations of each chapter should only be considered as those of the contractor when the task report was written.

This development report is intended to compile the data which, with the addition of EPA analysis, formed the basis for the EPA Dynamometer Specification. As an example of the EPA analysis, the EPA has determined that motoring ability of the dynamometer would greatly facilitate the calibration procedure, and is, therefore, desirable for the EPA dynamometer. The addition of this requirement may significantly alter the relative ranking of the dynamometer systems considered in Task 7. The Dynamometer Specification should be considred as a reference for the current views of the EPA motorcycle dynamometer requirements.

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Section 1

TASK 1 - REVIEW OF REQUIREMENTS

1.1 INTRODUCTION

The objective of this phase of the program is the development of specifications for a motorcycle dynamometer and motorcycle cooling system to be utilized in emission certification programs. The specifications must address all the attendant problems of dynamically simulating vehicle road conditions. In the development of dynamometer specifications, various power absorbers, roll assemblies and inertia assemblies will be evaluated and related to actual road-load data. Variable-flow blower systems which simulate actual engine cooling effects will be examined. Specific parameters to be studied include blower style, ducting requirements, noise levels, efficiency, power requirements, flow control methods, cost, and delivery. In addition to establishing the specifications for the dynamometer and the cooling system, the general test cell layout, the requirements for utilities, and the controls for the test cell's environment will be examined.

1.2 TASK 1 - REPORT OBJECTIVES

This section will summarize the pertinent requirements and examine the effect these will have on the design of the dynamometer, the cooling system and auxiliary systems.

Where possible, general and preliminary specifications will be presented and general design criteria will be examined.

1.3 REVIEW OF REQUIREMENTS

The EPA's "Proposed Rulemaking (NPRM) for New Motorcycles," dated 22 October 1975, and the system requirements, as outlined in the Request for Proposal and contract, were reviewed. Relevant requirements are listed below, and their impact on the design and specification for the motorcycle dynamometer and cooling system are discussed.

1.3.1 Definition

A motorcycle is defined in the NPRM as any motor vehicle designed to operate on not more than three wheels in contact with the ground which is not a passenger car or passenger car derivative. Additionally, for the purpose of testing, the engine displacement must be 50 cc or larger, and the cycle will be street legal. This definition specifically includes three-wheeled cycles. The implications of designing a facility for testing for both two- and three-wheeled motorcycles will be discussed in Section 1.6.

1.3.2 Test Cell Environment

The NPRM specifies that all test phases be conducted with an ambient temperature range between 20°C and 30°C. These temperature limitations apply to both the test cell itself and any adjoining space where the emission analysis might be conducted.

No specific requirements are placed on humidity or air contaminant levels within this facility; but the control of these parameters is important. Reasonable humidity control will be achieved by adequately designing the heating/airconditioning/ventilation system. Background air contaminant levels must also be maintained at low levels for reasons of work safety and health and to provide reasonable background levels emissions analysis.

1.3.3 Emission Test Equipment

The scope of this program precludes the development or specification of the emission test equipment. However, because this equipment is to be placed in the facility, space, utility and interface requirements for the equipment must be considered. The following equipment may be housed in the dynamometer cell and test facility:

- Constant Volume Sampler (CVS)
- Dilution air filter assembly
- Tail pipe connector and flexible tubing
- Exhaust gas analytical system housing nondispersive infrared analyzers for measurements of carbon dioxide and carbon monoxide, a chemiluminescent NO_x analyzer and an FID hydrocarbon analyzer.
- Storage facility for 20 gas cylinders used in the operation and calibration of the analytical system.

1.3.4 Motorcycle Dynamometer

The dynamometer requirements are discussed in Section 85.478-15 of the NPRM and the Request for Proposal.

These documents discuss the requirement for simulating motorcycle inertia in 10 kg increments. The NPRM also calls out a maximum inertial mass of 480 kg. However, as required by the contract, overcapacity will be provided to a maximum of 700 kg. The design of this assembly will be developed in Task 4 of this report.

The NPRM does not specify the power absorption unit or the relationship between load and vehicle speed. It does define the road-load settings at 65 kph as a function of vehicle mass and procedures for determining road-load settings. The contract requires study and definition of the power absorption system. This will be performed in Task 2.

The dynamometer roll configuration will be defined in Task 3. However, the implications of testing three-wheeled cycles will be discussed in Section 1.6 of this report. The only other reference to the dynamometer in the NPRM concerns the attitude of the motorcycle on the dynamometer; and it is specified that the bike be level when tested.

1.3.5 Engine Cooling System

The contract and the test procedure described in the NPRM requires, a variable-speed blower system to simulate engine cooling. According to the contract, at roll speeds in excess of 10 kph, the duct exit velocity from the blower should be within 10 percent of the corresponding roll speed, and at roll speeds less than 10 kph, the air velocity should be within ±1 kph. Also, the contract calls out an outlet area of 0.5 m^2. Additionally, the regulations specify that the blower outlet will be between 0.15m and 0.2m above floor level and that the blower outlet will be squarely positioned between 0.3m and 0.45m in front of the vehicle's front wheel.

The evaluation of blowers and fans for this application will be presented in the review of Task 5 activities. In this report, general specifications regarding the blower's power requirements and positioning in the cell will be presented. Additionally, the impact of utilizing the same blower system for emissions tests of three-wheeled vehicles and two-wheeled cycles will be examined.

1.3.6 Driving Cycle for Emission Test

Two driving cycles to be utilized in the emissions test are defined. One is applicable to the smaller motorcycles, engine displacement less than 170 cc, and the other to motorcycles with engine displacement 170 cc or greater. The driving cycle for the larger vehicles requires equivalent accelerations and decelerations, but higher maximum velocities. These velocities, accelerations, and decelerations will have to be matched by the blower system for simulating engine cooling. Maximum speed encountered during the test cycle is 90.1 kph (25.0 m/sec). The maximum sustained acceleration occurs between 447 seconds of the cycle and 455 seconds when the acceleration rate is 5.3 km/hr/sec (1.47 m/sec^2). The maximum deceleration rate is 5.3 km/hr/sec (1.47 m/sec^2).

1.3.7 Driving Cycle for Mileage Accumulation

Mileage accumulation procedures are detailed in the NPRM for motorcycles having an engine displacement less than 170 cc and greater than 170 cc. Both the mileage accumulation cycle and total amount of mileage accumulated differs for the two engine classifications. The larger motorcycles are subject to a more demanding test, requiring an accumulation of 30,000 km at speeds reaching a maximum of 110 kph (30.5 m/sec).

A dynamometer designed for mileage accumulation test will require increased blower and power absorption capacities. Additional safety precautions also must be considered. All of these implications are discussed in Section 1.5. In this report, only a dynamometer for emission testing will be discussed in detail.

1.4 SAFETY CONSIDERATIONS

A review of OSHA regulations and the California Industrial Safety Code was conducted to determine applicable regulations. Few of the provisions of these regulations apply to the dynamometer testing of motorcycles. Those that do apply will be examined below. In addition, safety considerations peculiar to the testing of motorcycles also must be examined.

The following sections of the OSHA regulations affect the design of the dynamometer facility:

- 1910.36 (b) (8) requires two exit paths from the area.
- 1910.95 defines limits of noise exposure without supplementary protection. Noise exposure limits are listed below.

Duration per Day (hr)	Sound Level (dBA)
8.0	90
6.0	92
4.0	95
3.0	97
2.0	100
1.5	102
1.0	105
0.5	110
0.25	115

New regulations under consideration may reduce these exposure levels and require the installation of noise monitors.

- 1910.101 describes procedures for the storage of compressed gases. These procedures have been developed by the Compressed Gas Association.

- 1910.157 defines the number and type of fire extinguishers required and their location.

Of the OSHA regulations, noise control will be the most difficult to satisfy. Noise will emanate from the blower and the motorcycle engine and drive train. Reverberation reduction by the use of acoustical cell walls will be helpful and noise will be a parameter considered in the selection of the blower system. The use of ear protectors may be mandatory. Eye protection also should be considered because of the possibility of particles or small objects being carried in the cooling airstream.

The possible consequences of mechanical failures of the motorcycle in terms of operator injury must be evaluated. Such things are rear wheel lockup, tire failure or drive chain failure, and fuel tank failure should be anticipated and appropriate operator safeguards be incorporated in the design of the test facility.

In the event of such failures, it is appropriate to perform emergency shutdown operations automatically, such as stopping the vehicle engine, stopping the cooling air, shutting off the fuel and bringing the rolls to stop, all without operator action. Emergency buttons also should be provided by which the operator and/or test observer can initiate emergency shutdown in case automatic detection devices fail to perform or a malfunction occurs which is not sensed by the automatic devices.

1.5 SYSTEM ADAPTABILITY FOR MILEAGE ACCUMULATION TESTS

As discussed in Section 1.3, the mileage accumulation test procedure subjects the motorcycle to higher velocities and accelerations than occur in the emissions certification driving cycle. Therefore, greater loads and operating ranges will be required if the dynamometer facility is to be utilized in both modes. Specifically, in the mileage accumulation cycle, the dynamometer will have to absorb twice the road-load experienced in the emissions driving cycle. In order to simulate engine cooling conditions during the mileage accumulation test, the linear velocity at the exit of the blower duct must be 30.5 m/sec., 20 percent greater than for the emissions' cycle and requiring a 25 percent larger motor, in terms of horsepower.

While it is a simple matter to equip the dynamometer and cooling system with components to meet the requirements imposed by both test modes, the human safety factor becomes significantly more critical during the mileage accumulation test. Over 1,000 hours will be required to accumulate 30,000 km, versus less than 1 hour to complete an emissions test. It can be assumed that component failure is more likely to occur in the process of mileage accumulation than during emissions measurement. In addition, driver fatigue will increase safety risks and the likelihood of errors.

1.6 SYSTEM ADAPTABILITY FOR TESTING THREE-WHEELED MOTORCYCLES

The NPRM applies to three-wheeled motorcycles as well as two-wheeled bikes. Therefore, general design parameters which will be affected by the inclusion of these vehicles must be examined.

The design modifications necessary to test both two- and three-wheeled motorcycles on a chassis dynamometer are:

- Provision for added roller or rollers to engage the two drive wheels that will span the minimum and maximum tread dimensions.
- Realignment of cooling air discharge nozzle.
- Relocation of vehicle restraint system (e.g., front wheel clamp or cables).
- Relocation of driver's aid and controls.
- Means for disconnecting added roller to reduce windage, friction and inertia when testing two-wheeled motorcycles.
- Increased test cell and pit width.
- More extensive protective covers or floor plates.

In light of these design problems and the resultant design complexities, this report will limit itself to the examination of a suitable design for performing emissions measurement on two-wheeled motorcycles.

Section 2

TASK 2 - EVALUATION OF DYNAMOMETER POWER ABSORBERS

2.1 INTRODUCTION

This task report will present an analysis of power absorption units (PAU's) and PAU control systems with respect to their suitability for use in duplicating motorcycle road-loads in performing exhaust emissions test procedures, as defined in the Notice of Proposed Rulemaking (NPRM) and EPA Contract No. 68-03-2141. These requirements will be examined and a general specification for the PAU and control will be presented. A detailed analysis of PAU's and controls is presented in this task report, and recommendations are made.

2.2 REVIEW OF REQUIREMENTS

The NPRM defines the driving cycle for emissions measurement in Appendix I and the durability driving schedule in Appendix IV. The EPA project officer has indicated that emissions measurements are of primary importance in the design of the dynamometer, and the durability cycle is to be considered as an added capability if found to be compatible with the requirements.

Both procedures require the motorcycle to be operated over a prescribed driving cycle which includes idle, steady-state cruise, and acceleration and deceleration modes. The NPRM provides road horsepower loads for motorcycles according to classes of loaded vehicle mass at one speed, 65 kph. This is analagous to the procedures employed in the testing of light-duty vehicles. In automobile testing, the usual practice has been to provide the inertia load component by means of roller-driven flywheels and the road-load component by means of a roller-driven PAU having a near cubic/load characteristic.

2.3 TERMINOLOGY

The following terms and symbols are used in the discussion of PAU's and control systems.

Power Absorption Unit (PAU): A device which applies torque to the roller on which the motorcycle drive wheel bears. (A PAU in combination with means for measurement of torque and rpm is usually termed a dynamometer.)

Kilowatt (kw): Unit of power adopted by Systems International (SI). One U.S. Horsepower (550 ft. lbf/sec) equals 0.7457 kw. One metric horsepower (HPm) equals 0.7355 kw.

2.4 LOAD DETERMINATIONS

As part of this contract, the EPA has supplied the contractor with empirical road-load data for several motorcycles, expressed in the form of an equation which defines the force required to move a motorcycle at any constant speed. This equation and the empirical load factors for several motorcycles are shown in Table 2-1. The relative influence of each of these terms of the quadratic equation is displayed in Figure 2-1 for three motorcycles.

Using the formulae summarized in Table 2-1, further calculations were made to establish the road-load horsepower

TABLE 2-1

EPA MOTORCYCLE PERFORMANCE DATA

(Received March 3, 1975)

$F = F_0 + F_1 V + F_2 V^2$ where

V = meters/sec.
mass = kilograms
F = Newtons

NOTE: F_0 to mass correlation coefficient is 0.92
Assume mass = inertia; wheels, etc. neglected

MODEL	MASS	F_0	F_1	F_2
Suzuki 750	345	57.3	0.0	0.412
H.D.	395	54.3	2.02	0.362
Suzuki 50	159	23.3	.793	0.281
Yam. Enduro	195	21.2	1.76	0.300
Honda CT 70	135	24.3	.609	0.234
Yam 125 Rd	205	28.2	1.50	0.303
Kaw 175	195	24.9	1.98	0.276
Honda 360	245	26.0	2.51	0.265

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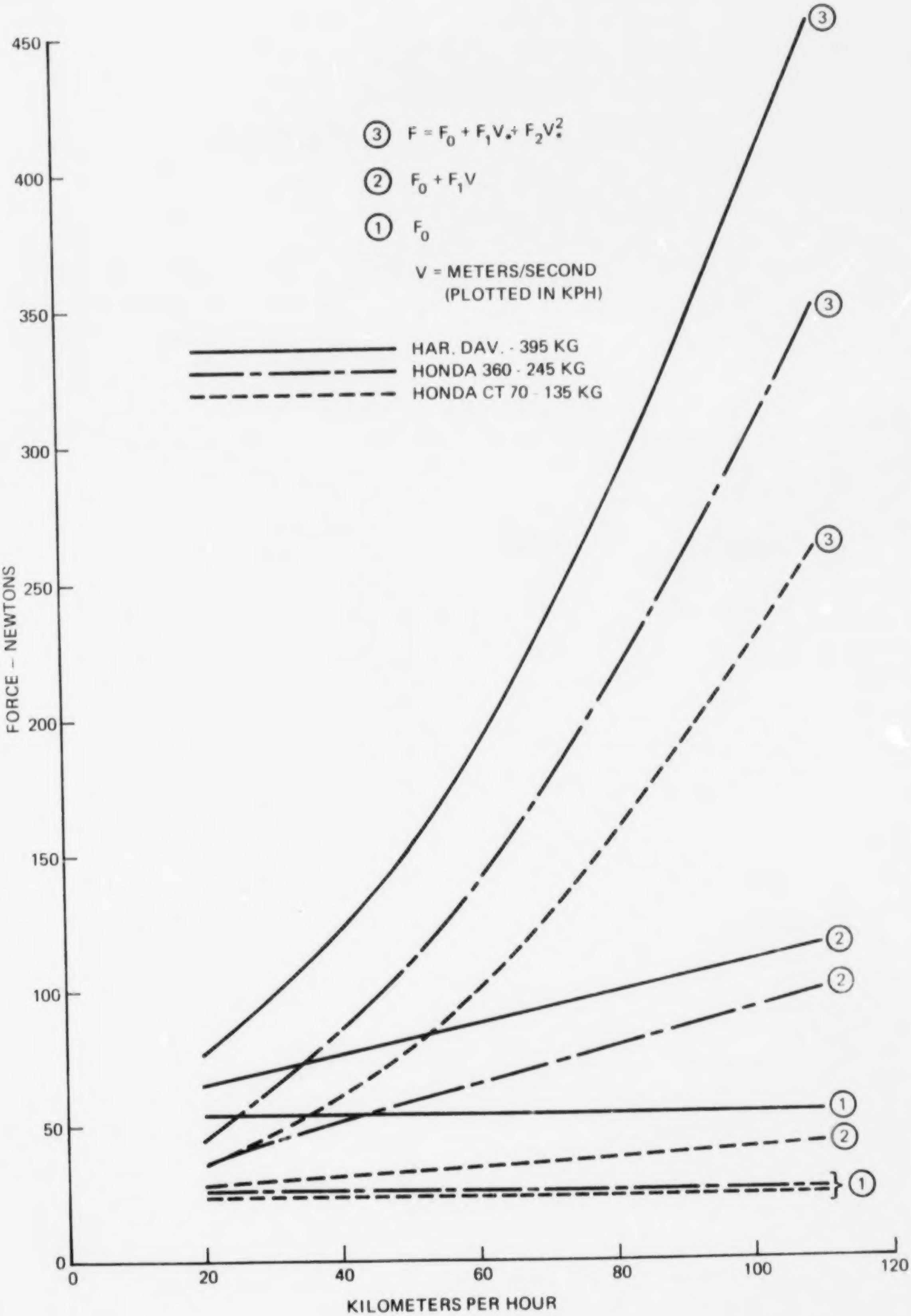


Figure 2-1. Motorcycle Road Load Force

levels as a function of speed for each motorcycle. These data have been listed in Table 2-2. For each set of horsepower/speed data a best fit power curve in the form of:

$$Y = kx^c \tag{2-1}$$

where Y = watts
x = speed (meters/second)
c = power coefficient
k = constant

was calculated for comparison with the existing load curve for light-duty vehicle dynamometer, and this is also tabulated in Table 2-2. From these data for the eight motorcycles, a mean exponent was established and found to be 2.18 with a standard deviation of 0.08 and a standard error of 0.29. As a comparison, the mean exponent used to describe the performance of the Clayton dynamometer used to test light-duty vehicles is between 2.83 and 3.0. Results of tire-to-roller losses, discussed in Section 3, combined with the road-load horsepower predicted by the EPA empirical data are presented in Figures 2-2 and 2-3. The predicted road-load, tire-to-roller loss and net PAU horsepowers are plotted. These data indicate that there is a considerable difference in the exponential characteristic of the required PAU load between the Honda 90 cc and Harley Davidson 1200 cc motorcycles.

2.5 GENERAL SPECIFICATIONS

Several factors have been considered in preparation of general specifications for the PAU. These are:

- Power capacity required primarily for emissions testing and secondly for durability driving.

TABLE 2-2

MOTORCYCLE ROAD LOAD REQUIREMENTS

								KILOMETERS PER HOUR			
MODEL	MASS	F_o	F_1	F_2	20	40	50	65	85	110	
SUZUKI 750	345	57.3	0.0	0.412	.529	1.63	2.58	4.70	9.21	18.36	HP
					70.0	108.2	136.8	191.6	287.0	442.0	F
	$Y = (8.76 \times 10^{-4})\ x^{2.08}$				0.0	0.0	0.0	0.0	0.0	0.0	F_1V
					12.7	50.9	79.5	134.3	229.7	384.7	F_2V^2
					57.3	57.3	57.3	57.3	57.3	57.3	$F_o + F_1V$
HAR. DAV.	395	54.3	2.02	0.362	.579	1.83	2.87	5.13	9.75	18.86	HP
					76.7	121.4	152.2	208.8	303.8	454.0	F
	$Y = (1.11 \times 10^{-3})\ x^{2.04}$				11.2	22.4	28.1	36.5	47.7	61.7	F_1V
					11.2	44.7	69.8	118.0	201.8	338.0	F_2V^2
					65.5	76.7	82.4	90.8	102.0	116.0	$F_o + F_1V_1$
SUZUKI 50	159	23.3	.793	0.281	.275	1.01	1.67	3.17	6.38	12.87	HP
					36.4	66.8	88.5	129.2	198.7	309.9	F
	$Y = (2.75 \times 10^{-4})\ x^{2.26}$				4.4	8.8	11.0	14.3	18.7	24.2	F_1V
					8.7	34.7	54.2	91.6	156.7	262.4	F_2V^2
					27.7	32.1	34.3	37.6	42.0	47.5	$F_o + F_1V$
YAM. ENDURO	195	21.2	1.76	0.300	.304	1.18	1.95	3.70	7.38	14.75	HP
					40.2	77.8	103.5	150.8	230.0	355.1	F
	$Y = (2.93 \times 10^{-4})\ x^{2.28}$				9.8	19.6	24.4	31.8	41.6	53.8	F_1V
					9.3	37.0	57.9	97.8	167.2	280.1	F_2V^2
					31.0	40.8	45.6	53.0	62.8	75.0	$F_o + F_1V$

TABLE 2-2

MOTORCYCLE ROAD LOAD REQUIREMENTS (Continued)

							KILOMETERS PER HOUR				
MODEL	MASS	F_o	F_1	F_2	20	40	50	65	85	110	
HONDA CT 70	135	24.3	.609	0.234	.264	.906	1.47	2.74	5.43	10.86	HP
					34.9	60.0	77.9	111.6	169.1	261.4	F
	$Y = (3.30 \times 10^{-4}) \times^{2.18}$				3.4	6.8	8.5	11.0	14.4	18.6	F_1V
					7.2	28.9	45.1	76.3	130.5	218.5	F_2V^2
					27.7	31.1	32.8	35.3	38.7	42.9	$F_o + F_1V$
YAM. 125 RD	205	28.2	1.50	0.303	.347	1.24	2.03	3.78	7.46	14.83	HP
					45.9	82.3	107.5	154.1	232.5	356.9	F
	$Y = (4.11 \times 10^{-4}) \times^{2.20}$				8.3	16.7	20.8	27.1	35.4	45.8	F_1V
					9.4	37.4	58.4	98.8	168.9	282.9	F_2V^2
					36.5	44.9	49.0	55.3	63.9	74.0	$F_o + F_1V$
KAW 175	195	24.9	1.98	0.276	.336	1.22	1.99	3.70	7.24	14.25	HP
					44.4	81.0	105.6	150.6	225.5	343.1	F
	$Y = (4.08 \times 10^{-4}) \times^{2.20}$				11.0	22.0	27.5	35.7	46.7	60.5	F_1V
					8.5	34.1	53.2	90.0	153.9	257.7	F_2V^2
					35.9	46.9	52.4	60.6	71.6	85.4	$F_o + F_1V$
HONDA 360	245	26.0	2.51	0.265	.341	1.26	2.06	3.80	7.38	14.42	HP
					45.1	83.6	109.0	154.7	230.0	347.1	F
	$Y = (4.21 \times 10^{-4}) \times^{2.20}$				13.9	27.9	34.9	45.3	59.3	76.7	F_1V
					8.2	32.7	51.1	86.4	147.7	247.4	F_2V^2
					36.9	50.9	57.9	68.3	82.3	99.7	$F_o + F_1V$

$Y = (4.546 \times 10^{-4}) \times^{2.18}$ $\quad 2 = .9641$

$\bar{x} = 2.18$ $\quad S = 0.0821$ $\quad S\ \bar{x} = 0.290$

Mean Std. Deviation Std. error

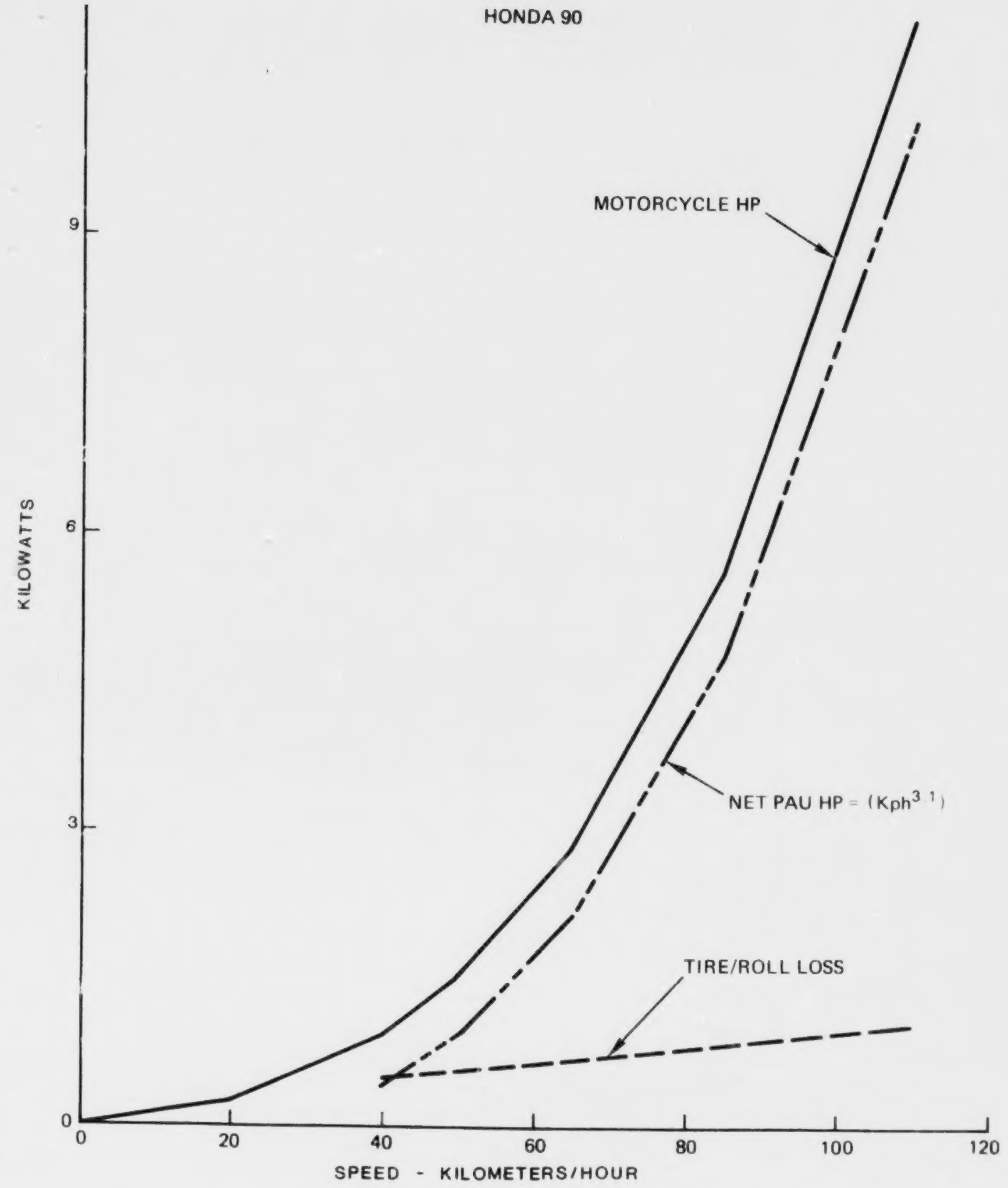


Figure 2-2. Net Road Load Determinations

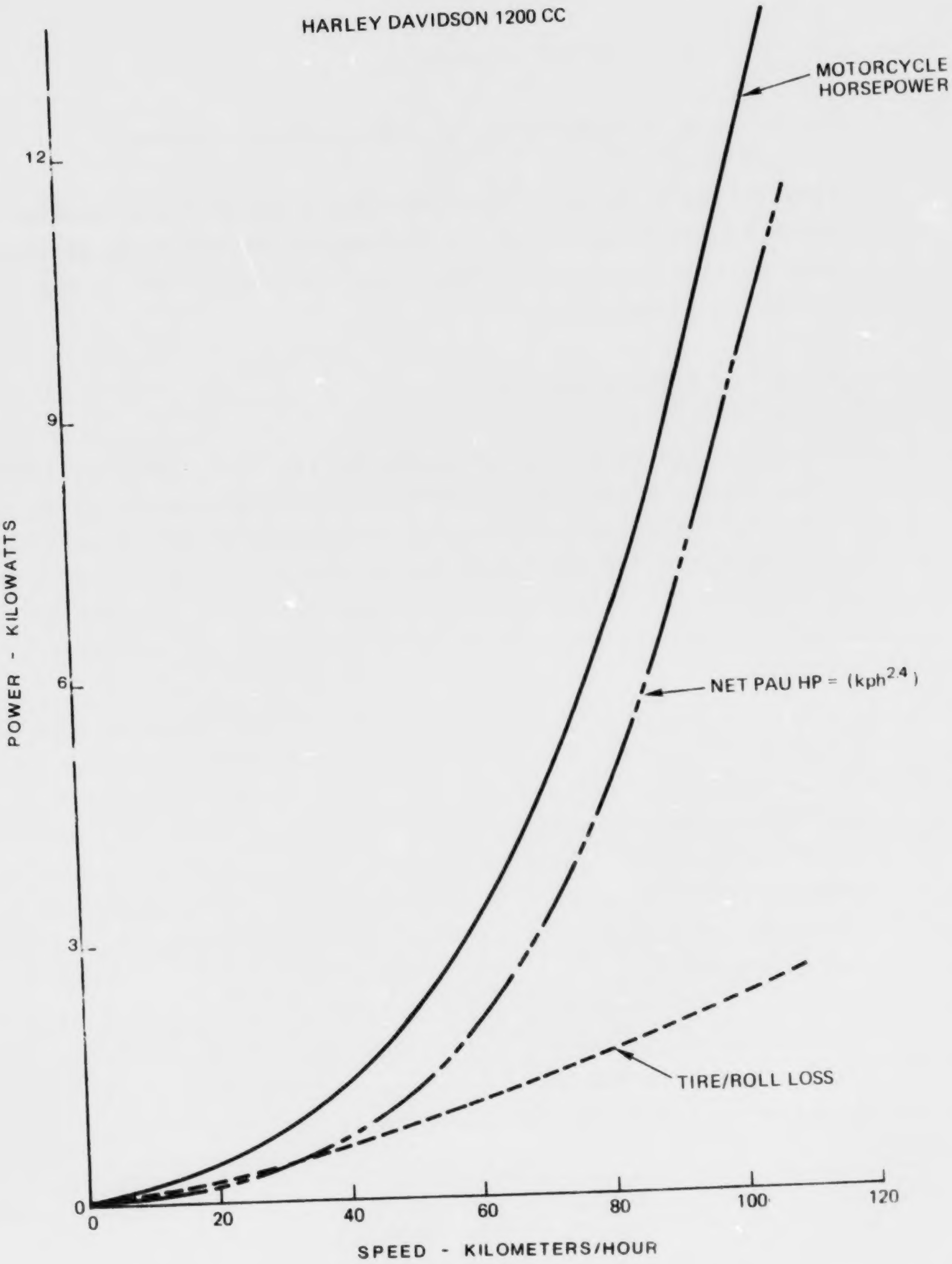


Figure 2-3. Net Road Load Determinations

- Inertia simulation method.
- Roller diameter.
- Complexity of PAU controls required.

Many of these factors interact, which necessitate making at least tentative engineering judgements to avoid an unmanageable selection matrix. These shall be discussed in the following paragraphs.

2.5.1 Power Capacity

The required PAU capacity for road-load, as calculated from the EPA formula and data on eight motorcycles is shown in Table 2-2. A maximum of about 14.7 kw (20 HPm) is indicated, including operation at 110 kph. Therefore, a PAU of 18.4 kw (25 HPm) absorption capacity would provide an adequate reserve to perform both emissions and durability cycles.

If the PAU is also to be used for inertia simulation, it must be capable of both motoring and absorbing, and the horsepower capacity must be increased according to the amount of inertia simulation. Maximum PAU inertia simulation, less the intrinsic inertia of the roller, PAU, etc., would result in about 37 kw (50 HPm) absorbing and 18.4 kw (25 HPm) motoring requirement if both emissions and durability cycles were considered.

The increased cost of the PAU for inertia simulation and the probable errors due to control and PAU response lag as discussed in the Task 4 report, are considered sufficient reasons to limit the PAU function to providing road-load.

2.5.2 Roller Diameter

The roller diameter influences the PAU specification because it determines the PAU rpm for a given equivalent road speed, assuming that the PAU is directly coupled to the roller shaft. The Task 3 report recommends a 530.5 mm diameter roller, which results in 1,100 rpm at 110 kph.

Most PAU's of the required power capacity are designed for operation at 4,000 to 6,000 rpm which could be achieved by means of a gear box or other type of speed increaser. However, it may not be possible to sufficiently reduce the losses in such a speed increaser to the point that the losses would be insignificant. Four PAU installations can be considered:

- Mount the gearbox on the nose of the PAU and trunnion mount the combined unit.
- Foot-mount both the PAU and the gearbox and connect the output shaft to the roller shaft by means of a shaft torquemeter.
- Provide corrections to the PAU command signal to compensate for predetermined gearbox losses.
- Select a PAU of a larger rated capacity, having sufficient torque load to absorb the required power at the lower RPM.

The use of an integral PAU and gearbox does not significantly degrade measurement accuracy. However, gearbox losses do limit the minimum torque of a nonmotoring PAU and necessitate operating a motoring PAU through the zero power point. The use of a foot-mounted PAU and gearbox with an output shaft torquemeter yields similar results. Compensation of the PAU control for gearbox losses is poor because the losses vary with temperature, rpm, and load. Therefore, it is felt that the selection of an overrated PAU which has sufficient torque load to absorb the required power at the lower rpm should be pursued. Information on PAU's which can absorb 18 kw at 1,100 rpm has been solicited from vendors and is discussed in Section 2.7.

2.5.3 PAU Controls

The controls required to produce loading according to an algorithm, such as that furnished by the EPA, must have the capability to accept the load factors F_0, F_1 and F_2 defined in Table 2-1. These are empirically determined values representing a constant, a linear and a square term in the formula for the road-load force for each individual motorcycle. The control must then calculate the instantaneous load force according to roller rpm and cause the PAU to apply this force (torque) to the roller shaft.

The tire-to-roller loss will have to be determined for multiple loads and tire tread constructions so that accurate corrections can be applied to the load command.

2.6 POWER ABSORPTION UNIT CONSIDERATIONS

In this section parameters which affect the performance and selection of a PAU are considered. These include:

- Utilities required
- Moment of inertia of rotor
- Method of heat dissipation
- Simulation of vehicle inertia

- Calibration
- Stability of calibration
- Maintenance requirements
- Adaptability to test cycle revision

Detailed information on the performance of specific PAU's is presented in Section 2.7.

2.6.1 Utilities Required

The discussion of utilities here is limited to those for operation of the PAU and control.

Cooling water is generally required for removing heat generated by power absorption in many of the PAU's considered. Exceptions are the air brake, dry gap-type eddy current absorber and DC motor/generator.

Water supply pressure of 200 to 350 KPa (30 to 50 psig) and volumetric rates, of 8.5 m^3/s per KW (0.1 gal/min/HPm), are generally required. In any of these units a small minimum flow of water is required even though no power is being absorbed. The cooling water may be discharged into the drain or it can be recovered and recirculated through a cooling tower. The choice is made based upon local codes, whether or not there is any existing cooling tower system and/or ecological factors.

2.6.2 Moment of Inertia

The moment of inertia of the PAU rotor is noted because it, in combination with the roller, shaft and couplings, constitutes the minimum equivalent vehicle weight which can be tested with proper inertia simulation. Conversely, it is of interest in determining the minimum flywheel size for mechanical inertia simulation or the motoring and absorbing power requirements for PAU simulation systems.

2.6.3 Thermal Dissipation Effects

The dissipation of PAU heat by means of cooling water has been discussed above. The Lear Siegler eddy current PAU is cooled by radiation and by convection air. This results in additional heat load on the system for the control of ambient temperature in the test facility. DC motor/generator type PAU's are usually regenerative systems in which most of the power absorbed is returned to the electric power lines. The heat developed in the armature and field coils is dissipated by means of forced airflow from self-contained blowers. Heat from this source is of little consequence.

2.6.4 Inertia Simulation

Simulation of vehicle inertia is discussed in Task 4. However, it is appropriate to indicate that if PAU inertia simulation is used, then the PAU must be capable of both motoring and absorbing modes of operation and the power capacity must be increased by as much as 100 percent.

The DC motor/generator, universal eddy current and hydraulic pump/motor types of PAU's, which can be used for vehicle inertia simulation, all require, even for fixed load control, control systems that are more complex than those for similar operation of hydrokinetic PAU's. The additional equipment necessary to perform PAU inertia simulation and/or load control per the EPA algorithm would consist of a small analog or digital computer to generate the command signals for the controller.

2.6.5 Calibration

Calibration of the PAU system should involve all the components in the chain, including the calibration arm, the PAU cradle bearings, the PAU to load cell linkage, the load cell cable and connectors, the load cell power supply, amplifiers, attenuator switches and readout instruments. As an example, the torque measuring system should be checked by the loading of certified weights on an arm extending from the PAU and observing the display on the torquemeter. Similarly, rpm should be checked by rotating the roller shaft, illuminated by a line frequency strobe lamp, and reading the rpm display. Calibration checks which employ the shunting of a strain gauge bridge, application of reference voltage or frequency to an rpm indicator, etc., are convenient and useful troubleshooting techniques and calibration checks, but are unacceptable for basic calibration.

The above examples are, perhaps, obvious. However, calibration of the mechanical or PAU-type inertia simulation system should be given particular attention. The technique and equipment for calibration of the system selected should be included in the final design.

2.6.6 Calibration Stability

Stability of calibration of the elements of the motorcycle dynamometer system is an important factor in the selection of components. Short-term drift in zero, span or linearity can be almost undetectable during a test run and can seriously degrade test-to-test correlation. It is, therefore, important to select system elements of known stability and immunity from temperature, vibration, supply voltage change, electrical "noise," etc. Special precautions should be taken to avoid interferences from noises emanating from high energy ignition systems.

2.6.7 Maintenance

The frequency and amount of maintenance, aside from calibration, determines the available operating time of the facility and thus, the operating cost to a large extent. This is believed to be particularly important when requirements for testing tend to be seasonal.

2.6.8 System Adaptability

The adaptability of the motorcycle dynamometer test facility to changes in the test cycle and the vehicles themselves is a valid design criteria. This influences the choice of PAU and control primarily with respect to the road-load characteristic. It is anticipated that fuel supply, environmental and consumer protection considerations will probably result in the need for increased precision of duplicating actual road-loading for emissions and fuel economy tests. Such requirements can be better met with a PAU and controller capable of loading the vehicle per the EPA algorithm in which the constant, linear and squared-load factors can be adjusted independently.

Unfortunately, the PAU systems having this capability are the most costly and longest in delivery time. The alternative, to improve delivery and reduce initial cost, would be to select a PAU which can meet present EPA requirements and can be upgraded by adding a more complex control system.

2.7 CHARACTERISTICS OF CANDIDATE PAU's

A basic dynamometer system is schematically shown in Figure 2-4. This system consists of a direct-driven PAU, a variable mechanical inertia system and a roller. The PAU

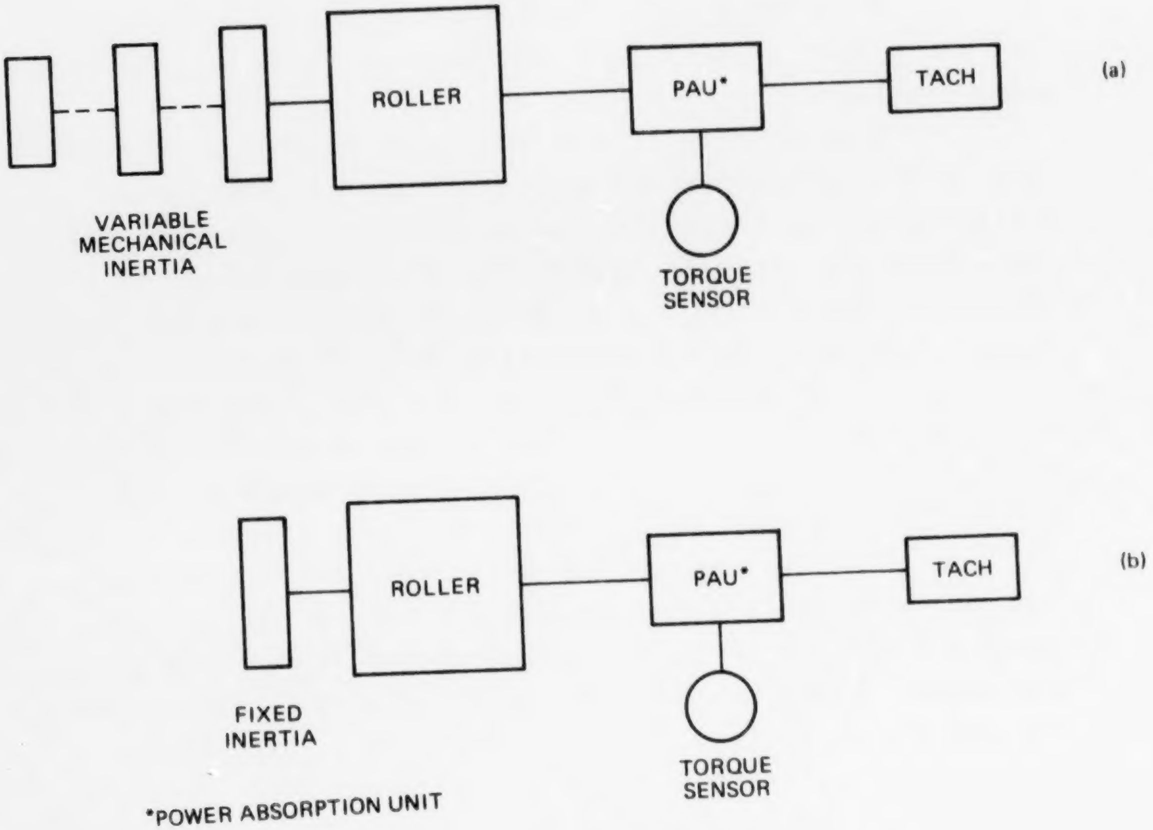

ROLLER
PAU*
TACH
(a)
VARIABLE MECHANICAL INERTIA
TORQUE SENSOR
ROLLER
PAU*
TACH
(b)
FIXED INERTIA
TORQUE SENSOR
*POWER ABSORPTION UNIT

Figure 2-4. Basic Dynamometer Systems

serves as the power absorber and the flywheel system simulates vehicle inertia. A second dynamometer system is also shown in Figure 2-4. It consists of a PAU, a roller, and a fixed flywheel. In this second system, the PAU serves two functions; power absorber and inertia simulator.

Some PAU's have fixed power absorption characterisitics. PAU's of this type include fans, blowers, hydraulic pumps and hydrokinetic units.

A more accurate duplication of actual road operation is possible by adding a control, such as that shown schematically in Figure 2-5, which develops the speed/load characteristic according to the EPA algorithm using the empirical load factors F_0, F_1 and F_2. The dynamometer control system in Figure 2-5 merely shows the elements required to operate the PAU on the EPA load algorithm. The empirical load factors F_0, F_1 and F_2 are manually set for the vehicle to be tested. Similar manual inputs are required for the vehicle mass and the tire/roll loss characteristic.

Roller speed and torque signals are fed back and compared with the program command. Errors or differences cause the PAU torque to be changed in the appropriate amount and sense. Other essential elements of the control system, such as signal filtering, proportional band, derivative function, etc., are not shown.

Inquiries regarding PAU characteristics were sent to representative groups of manufacturers. While some of the responses were limited, they were sufficient to permit an examination of candidate units. Specific types of PAU's described are:

- Air blower
- Friction brake
- Hydraulic pump
- Hydrokinetic - open flow
- Hydrokinetic - closed flow

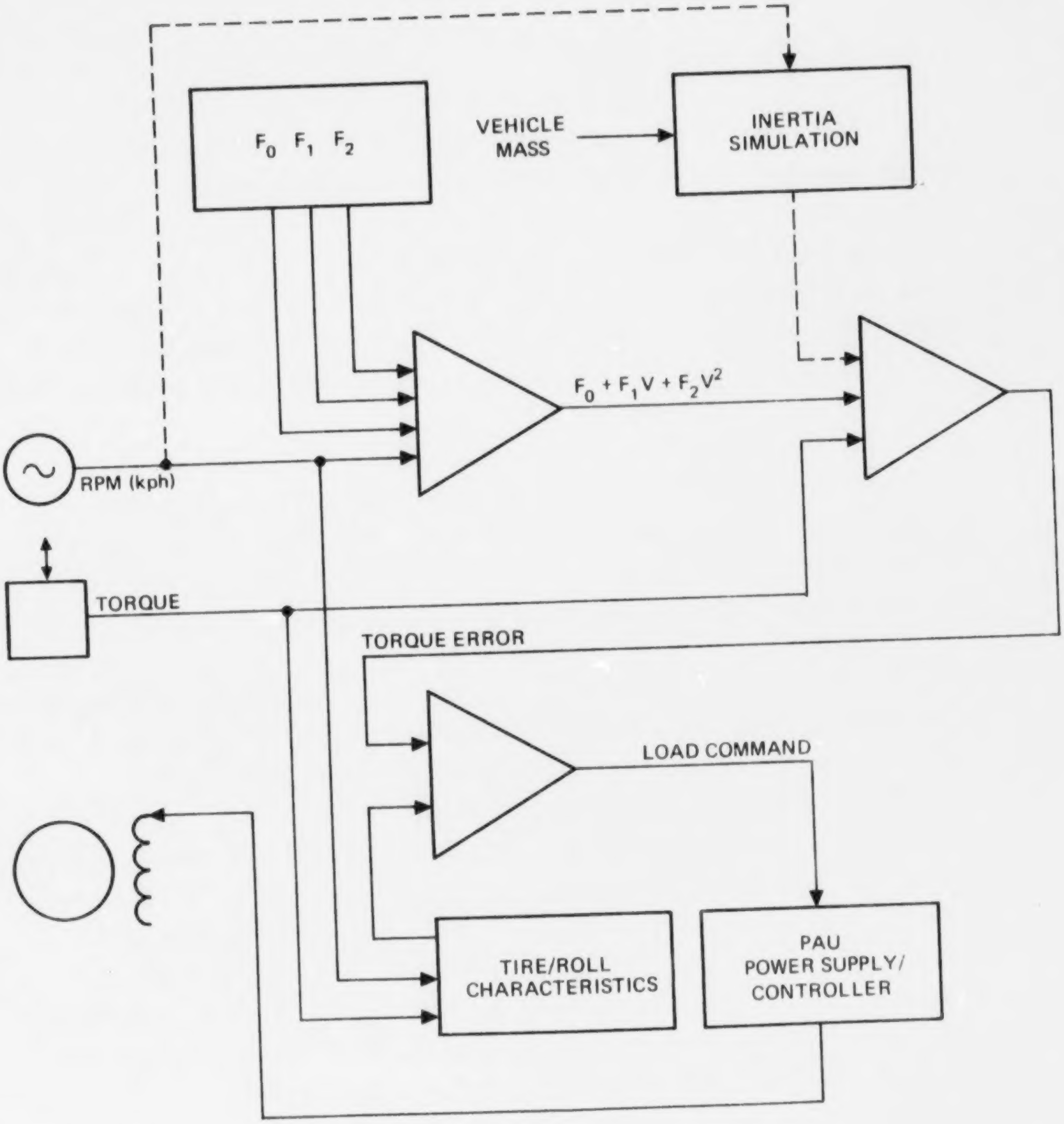

F0 F1 F2
VEHICLE MASS
INERTIA SIMULATION
F0 + F1V + F2V2
RPM (kph)
TORQUE
TORQUE ERROR
LOAD COMMAND
TIRE/ROLL CHARACTERISTICS
PAU POWER SUPPLY/ CONTROLLER

Figure 2-5. Dynamometer Control System

- Eddy current
- DC motor/generator
- AC adjustable speed drives

Their basic features are summarized in Table 2-3.

2.7.1 Air Blower

There are available several types of air blowers which exhibit an approximately cubic speed/load characteristic. Such a blower, driven by the roller upon which the motorcycle rear wheel rides, would appear to provide a simple low cost loading device for motorcycle testing. Unfortunately, there are several problems which render this type of PAU unsuitable for the accuracy and flexibility needed to perform the contemplated tests.

The horsepower speed characterisitc of an air blower is essentially fixed. It would be necessary to adjust the blower load to match the road-load characteristics of the motorcycle to be tested. This could be accomplished by throttling either the intake or discharge of the blower if it were directly driven from the roller shaft. Alternatively, an adjustable ratio drive could be interposed between the roller and the blower. Neither solution is believed to be practical. In addition, some type of damper control would be required to alter the blower's inherent cubic horsepower load curve in an effort to match the power curves cited in Table 2-2.

Adjustable dampers or orifices can be used only over a limited range without producing serious changes in the characteristic speed load curve of the blower. Further, they are susceptible to change in calibration with changes in air density due to temperature and barometric pressure variation.

TABLE 2-3

COMPARISON OF PAU SYSTEMS

	Utilities Required	Rotor Inertia kg-m2	Cooling Method	Vehicle Inertia Simulation By PAU	Complex Speed/Load Curve	Calibration Stability	Required Maintenance	Revision Adaptability	Approximate Price
Air Brake Typical	None	N.A. (Large)	None	No	No	Poor - Affected By Air Density	Low	No	N.A.
Hydraulic Pump Typical	Cooling Water, Control Electric Power	N.A. (Small)	Oil/Water Heat Exch.	Yes - Add Cost Of Hydraulic Power Supply	Yes - Add Electro - Hyd. Valve And Control	Fair - Depends On Fluid Temp. Control	Moderate - Chg. Hyd. Fluid And Filters	Good If Complex Control Type	N.A.
Hydrokinetic - open flow Go Power D-552	Cooling Water, Control Electric Power	3.954×10^{-2}	Water - Open Flow	No	No	Poor - Depends On Inlet/Outlet Flow	Low - Brgs. And Seals	No	$2 - 4K + Controls
Industrial Dyno	Cooling Water, Control Electric Power	1.171×10^{-2}	Water - Open Flow	No	No	Poor - Depends On Inlet/Outlet Flow	Low - Brgs. And Seals	No	$3.5K + Controls
Kahn Industries 61-124	Cooling Water, Control Electric Power	5.311×10^{-1}	Water - Open Flow	No	No	Poor - Depends On Inlet/Outlet Flow	Low - Brgs. And Seals	No	$7.3K + Controls
Hydrokinetic - closed flow Clayton Mfg. Co. D-18773	Cooling Water, Control Electric Power	2.221×10^{-2}	Water - Heat Exch.	No	Yes - Add PAU Fill/Drain System And Control	Good - Closed System	Low - Brgs. And Seals And Descale Ht. Exch.	Limited If Fixed Exponent Type, Good If Complex Control Type	$2.2K + Controls
Eddy Current Eaton Dynamatic	Cooling Water, Control And Excitation Electric Power	N.A.	Water - Dry Gap	Yes - Add Motor and Clutch	Yes - Add More Complex Control	Good - Depends On Closed Loop Control	Low - Brgs. And Seals And Descale	Good If Complex Control Type	$12 To 24K W/Controls
Lear Siegler C-40	Control and Excitation Electric Power	3.531×10^{-1}	Direct Radiation And Convection	No	Yes - Add More Complex Control	Good - Depends On Closed Loop Control	Low - Brgs. And Seals	Good If Complex Control Type	$2100 + Controls
Meidensha Elect.	Cooling Water, Control And Excitation Electric Power	4.7×10^{-1}	Water In Gap	Yes - Add Motor and Clutch	Yes - Add More Complex Control	Good - Depends On Closed Loop Control	Low - Brgs. And Seals	Good If Complex Control Type	$15K W/Controls
DC Motor/Generator Meidensha Elect.	Control and PAU Rating Electric Power	N.A. (Med.)	Power Recovery	Yes	Yes - Add More Complex Control	Good - Depends On Closed Loop Control	Moderate - Brushes And Bearings And Commutator	Good If Complex Control Type	$25K W/Controls
Reliance Electric	Control and PAU Rating Electric Power	2.528×10^{-1}	Power Recovery	Yes	Yes - Add More Complex Control	Good - Depends On Closed Loop Control	Moderate - Brushes And Bearings And Commutator	Good If Complex Control Type	$20K W/Controls

An infinitely variable ratio drive of sufficient power capacity would be costly. Changing the ratio between the roller and the blower would also result in changing the inertia of the system.

Because of the relatively low roller speed, the size of a blower for direct drive may be excessive. A speed increasing belt, chain or gear drive to permit use of a smaller size blower would add cost, and modify the characteristic load curve of the system because of the losses in the drive. Compensation for changes and the other variables would probably require a moderately complex control system, which is not, to our knowledge, commercially available.

2.7.2 Friction Brake

The friction brake PAU is composed of two distinct parts. The first of these is the rolling gear involving the rolls, bearings, mechanical frame, which holds it all together; and the mechanical parts of the brake discs and brake pads. The second part is the control system which electronically and pneumatically controls the application of pressure to the pads in order to provide holding torque for the vehicle under test. Figure 2-6 is a block diagram of the control system. It can be seen that the control system consists of two parts: first, an electronic part which calculates the desired torque from the speed signal generated by the dynamometer tachometer and then produces an electronic signal corresponding to the error between the calculated and measured torque; and secondly, a mechanical system which converts this electronic error signal into an air pressure command to the disc brake.

The mechanical system is composed of a solenoid which is driven by the current output of the electrical subsystem. The force on the plunger of this solenoid is proportional to the current flowing through the solenoid.

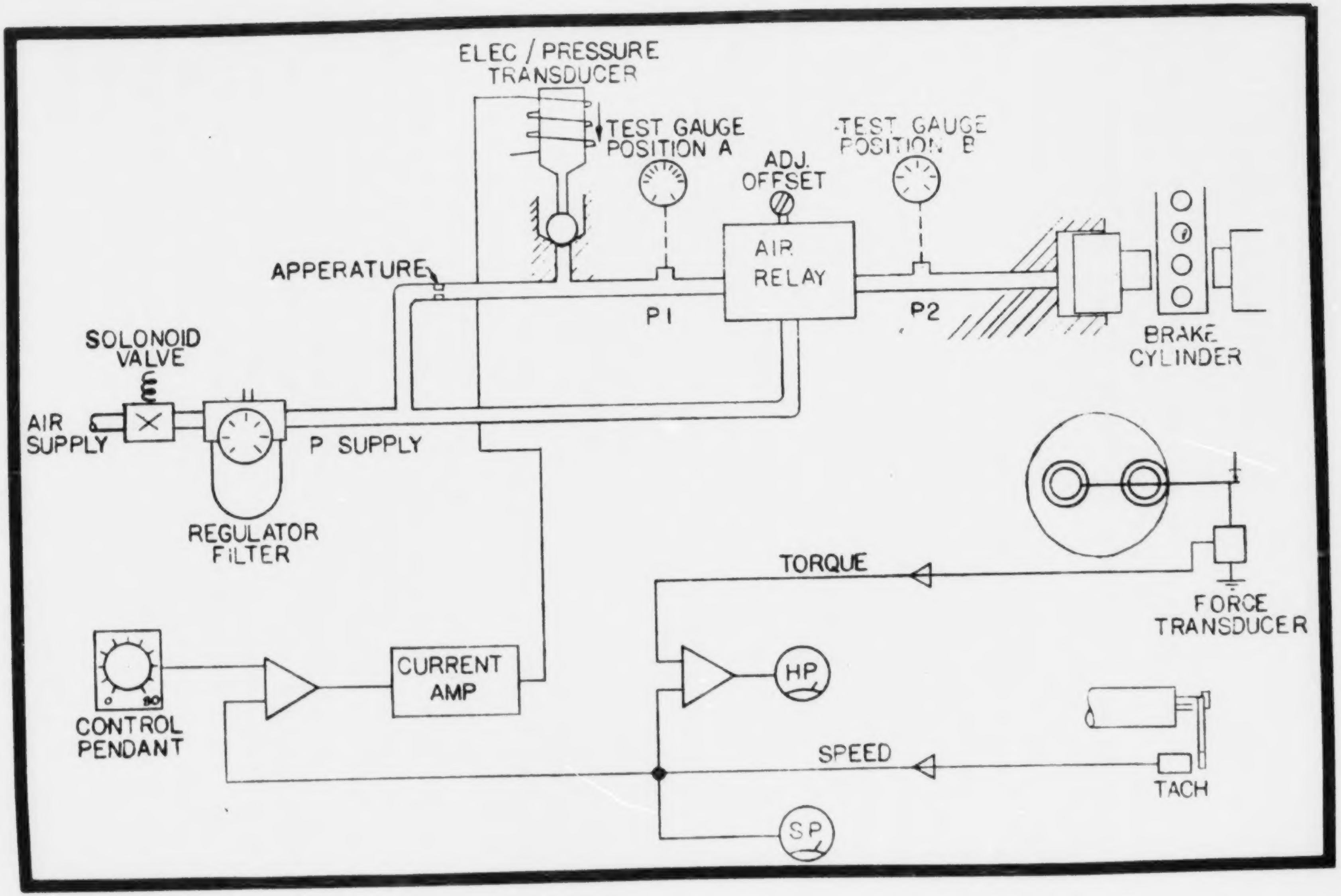

ELEC / PRESSURE TRANSDUCER
TEST GAUGE POSITION A
ADJ. OFFSET
TEST GAUGE POSITION B
AIR RELAY
APPERATURE
P1
P2
BRAKE CYLINDER
SOLONOID VALVE
AIR SUPPLY
P SUPPLY
REGULATOR FILTER
TORQUE
FORCE TRANSDUCER
CONTROL PENDANT
0
80
CURRENT AMP
HP
SPEED
TACH
SP

FIGURE 2-6. BLOCK DIAGRAM OF CONTROL SYSTEM

However, the constant of proportionality between the mechanical force and the current flowing depends on the position of the plunger in the solenoid and is variable, to some extent, from one unit to another.

The force of the plunger presses down on the top of a small ball. The pressure below the ball builds up until the air pressure on the bottom of the ball exceeds the mechanical pressure from the solenoid on the top of the ball. At this point, the ball rises off the seat and the air leaks out of the system at a rate rapid enough to maintain equilibrium. Thus, at all times, the mechanical pressure on the ball from above is approximately equal to the air pressure on the ball from the bottom. In order to compensate for pressure differences from 0 to about 414 kPa (60 psi), the ball needs to rise only a few thousands of an inch off the seat in order to provide the correct leakage rate.

Since the air for the electrical-to-pressure transducer is supplied through a very tiny orifice, the time required to build up the pressure or decrease the pressure in the electrical-to-pressure transducer would be very great if it had to drive any substantial volume. For this reason the output of the electrical-to-pressure transducer is applied to the control input of an air relay. The air relay has the ability to control a very large flow of air into the pistons.

The output of the air relay drives the four pistons on the brake assembly. The air relay is capable of both increasing or decreasing the pressure corresponding to changes in the inlet control pressure.

Intrinsically, friction brakes of the type described result in linear load curves. However, Autoscan Inc., has developed a Road Profile Simulator which allows the friction brake to be programmed to match almost any shape of load curve. An analog computer is used in conjunction with the algorithm to generate a voltage signal which is proportional to torque. A sensor measures the torque being applied to the brake and the two signals are compared, resulting in a corrective error signal. Autoscan has produced units which have selectable power curves. They have not developed a unit which can accept algorithm constants, though this can be developed.

One problem which has been experienced in previously developed units is control loop instabilities. This principally occurs when the vehicle is subjected to prolonged and pronounced accelerations and decelerations. In conversations with Autoscan personnel they admitted that their unit had not been used in conjunction with driving cycles and could not match the performance and versatility of an eddy current unit especially with respect to transient response characteristics.

The cost of the friction brake is approximately $3000-$4000. However, some development costs would be incurred in the design of a versatile system which could accept the EPA algorithm.

2.7.3 Hydraulic Pump

A hydraulic pump driven from the roller and discharging fluid through a variable orifice is a simple means of providing vehicle load. This system is used in the Hartzell Mark II motorcycle dynamometer and other commercially-available tuneup dynamometers. With this unit, pump discharge pressure and roll speed are used to obtain horsepower from a nomograph. Horsepower could also be calculated automatically from these data if the pump characteristics were known. Better accuracy could be obtained using pump reaction torque and roll speed directly. This basic system could be modified by the addition of a shaft or pump reaction torquemeter and an electrically actuated relief valve. With the addition of the control in Figure 2-5, it could load the roller per the EPA algorithm.

In the event that a nonflywheel inertia system is selected, the hydraulic system would become considerably more complex. First, the pump would have to be of a type capable of motoring, which complicates the control system. Second, motoring would require the addition of a hydraulic power supply.

Perhaps the best features of the hydraulic pump/motor are its inherently low inertia and small physical size for a given power capacity which make a high rate of response achievable.

Disadvantages of the hydraulic dynamometer system are:

- Potential safety hazard is somewhat higher than that of hydrokinetic or electrical-type systems due to presence of hot, high pressure oil.
- Longer warm-up time is required to reach the oil temperature and viscosity necessary for proper response of control system.
- Additional controls are needed to ensure stable temperatures within hydraulic system.
- The ancillary equipment such as the oil reservoir, heat exchanger, control valves, etc., require considerable space.
- Hydraulic systems tend to be sources of noise.
- Sophisticated hydraulic systems, as described above, have not been used before in this application and may require extensive development.

2.7.4 Hydrokinetic - Open Flow

The term "hydrokinetic - open flow" is used to describe power absorption units in which the working fluid, usually water, is introduced between a set of rotating and stationary discs within a circular housing. In these units which exhibit a nearly cubic speed/load characteristics, power is absorbed by the shear and turbulent flow of the water between the discs. The discs are perforated to increase the power absorbed in a given diameter and number of discs. These PAU's are "open flow" systems. The fluid is admitted into the housing near the axis of rotation and it collects at the periphery of this housing by centrifugal action. It is discharged at a controlled rate. The amount of housing fill and, thereby, the amount of power absorption is controlled by balancing the inlet and outlet flows. Precise control of supply pressure is essential for accurate and stable fill or load control. Water discharged from the PAU may be collected in a sump, cooled, deaerated and recirculated to conserve water, or it may be discharged into a drain. Schematics showing the operation of this type of PAU used in conjunction with manual or automatic load control systems are shown in Figure 2-7.

Hydrokinetic - open flow PAU's are relatively low cost in terms of horsepower capacity when the load is controlled manually. Because of the "open flow" operation, load index setting or polynomial speed/load operation require somewhat elaborate closed loop control systems as shown in Figure 2-7. The availability of control systems which can accept the EPA algorithm terms is unknown. It is only feasible to use hydrokinetic - open flow-type PAU's with mechanical inertia simulation because they lack motoring capability.

Go Power Systems furnished information on their "52" series absorption units, and their Model 552 has sufficient capacity for this application. Rotor inertia is

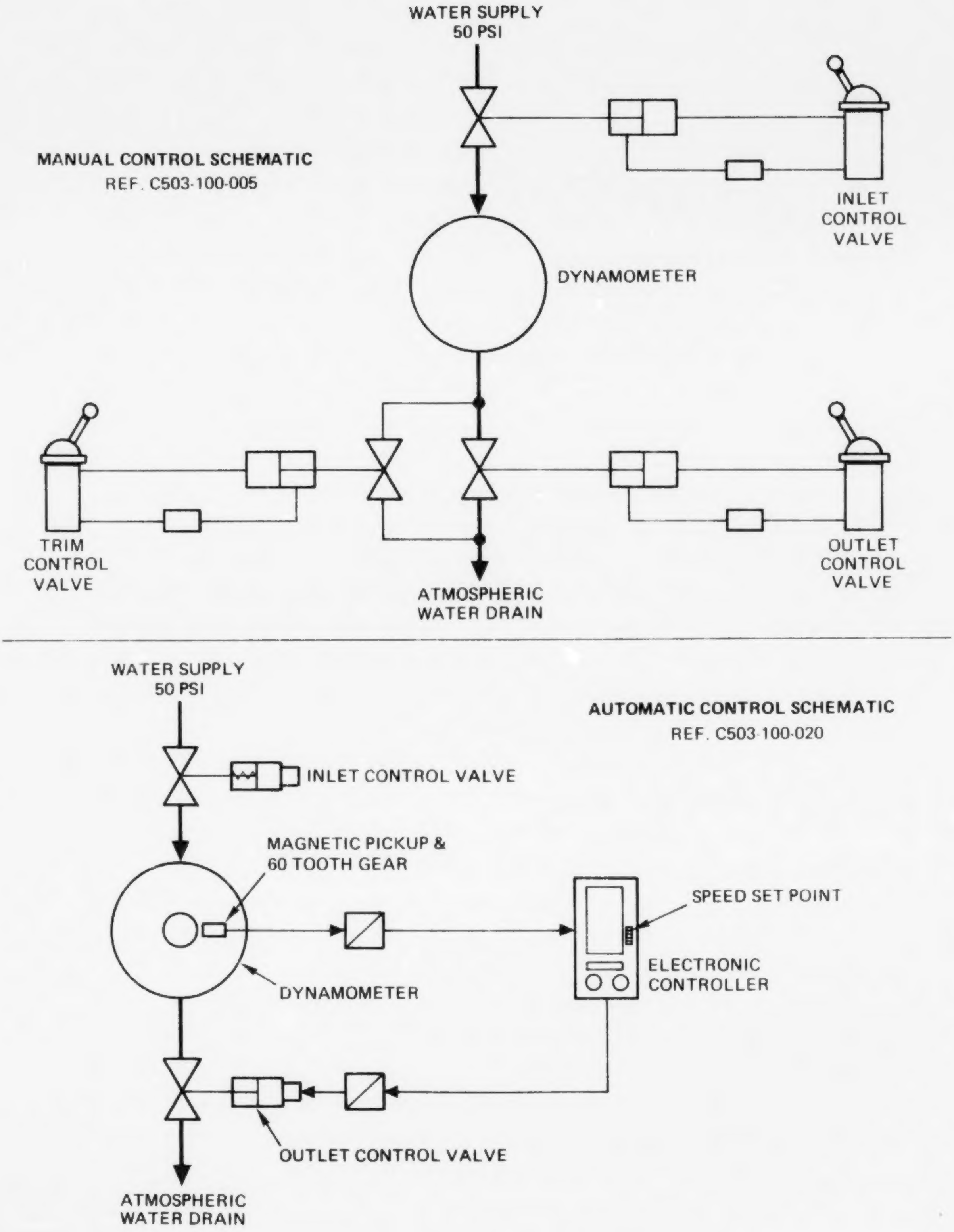

WATER SUPPLY
50 PSI
MANUAL CONTROL SCHEMATIC
REF. C503-100-005
INLET
CONTROL
VALVE
DYNAMOMETER
TRIM
CONTROL
VALVE
ATMOSPHERIC
WATER DRAIN
OUTLET
CONTROL
VALVE
WATER SUPPLY
50 PSI
AUTOMATIC CONTROL SCHEMATIC
REF. C503-100-020
INLET CONTROL VALVE
MAGNETIC PICKUP &
60 TOOTH GEAR
SPEED SET POINT
ELECTRONIC
CONTROLLER
DYNAMOMETER
OUTLET CONTROL VALVE
ATMOSPHERIC
WATER DRAIN

Figure 2-7. Hydrokinetic-Open Flow Control Systems

3.95×10^{-2} kg.m^2. Cooling water flow required is approximately 1.58×10^{-5} m^3/s at 37 kw (50 HPm). The price of the basic unit is "in the $2,000 to $4,000 range."

Industrial Dynamometer Company submitted product information, indicating that they would supply a PAU with a capacity of 37 kw (50 HPm). Water flow would be 1.89×10^{-4} m^3/s and the rotor inertia is 1.171×10^{-2} kg .m^2. Response time from zero to full load is stated to be approximately two seconds. The price of this unit is $3,500.

Kahn Industries, Inc., quoted their Model 061-124 for the motorcycle PAU application. The rotor inertia of this unit is 5.31×10^{-1} kg.m^2. Water flow of 1.85×10^{-4} m^3/s/kw is required. Kahn also supplied information on their manual and automatic load control systems. The price of the Kahn PAU is $7,250 and the automatic set point load control system is quoted at $4,877.

The hydrokinetic - open flow-type of PAU appears less than optimum for this application because of the need to develop a control system for maintaining a fixed exponential curve, other than cubic, or a complex speed/load equation.

2.7.5 Hydrokinetic - Closed Flow

The hydrokinetic closed flow type of PAU, exclusively built by Clayton Manufacturing, differs from the hydraulic friction type in that the working fluid moves spirally in a vaned toroidal chamber which is divided into a static section and a rotating section. The vanes are arranged so that the fluid is accelerated by the rotating half of the chamber and decelerated by the static half, thus developing reaction torque. A cross sectional view of this type of PAU is shown in Figure 2-8.

Reproduced from
best available copy.

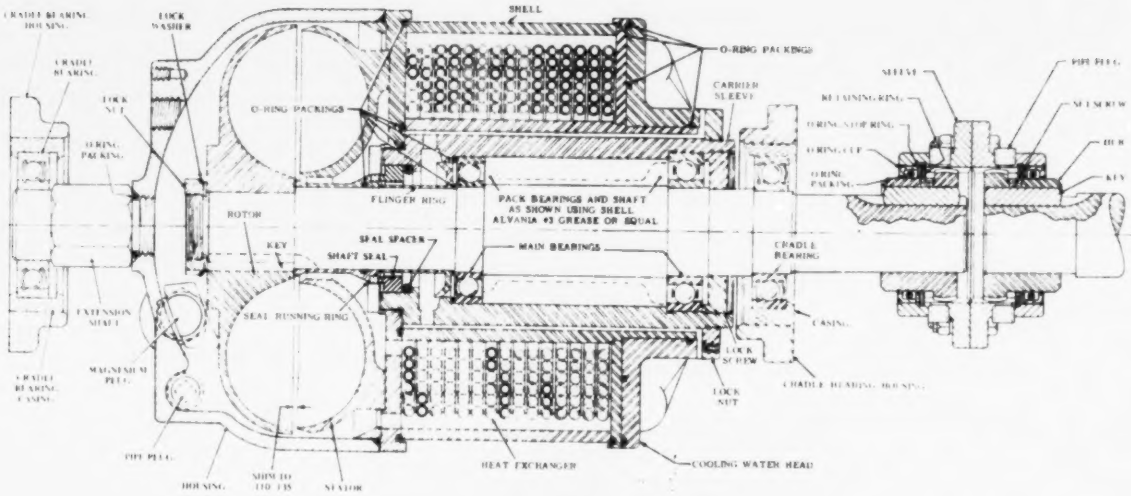

CRADLE BEARING HOUSING
LOCK WASHER
SHELL
O-RING PACKINGS
CRADLE BEARING
LOCK NUT
O-RING PACKING
O-RING PACKINGS
CARRIER SLEEVE
SLEEVE
PIPE PLUG
RETAINING RING
SET SCREW
O-RING STOP RING
HUB
O-RING CUP
O-RING PACKING
KEY
FLINGER RING
PACK BEARINGS AND SHAFT AS SHOWN USING SHELL ALVANIA #3 GREASE OR EQUAL
ROTOR
SEAL SPACER
KEY
SHAFT SEAL
MAIN BEARINGS
CRADLE BEARING
EXTENSION SHAFT
SEAL RUNNING RING
CASING
LOCK SCREW
MAGNESIUM PLUG
CRADLE BEARING CASING
LOCK NUT
CRADLE BEARING HOUSING
PIPE PLUG
HOUSING
SHIM TO 110-135
STATOR
HEAT EXCHANGER
COOLING WATER HEAD

Figure 2-8. Power Absorption Unit - Cross Section View

Heat generated in the working fluid is dissipated by means of a heat exchanger through which part of the toroidal flow is shunted. Thus, the need to precisely match open-flow hydraulic friction PAU's is eliminated. A schematic showing the operational components of this system is shown in Figure 2-9.

The torque developed at a given rotor rpm is directly related to the mass of fluid in the torus and the geometry of the vanes in both halves of the torus. The speed/load curve characteristic is controlled in design and manufacture to be slightly less than a cubic; actually 2.8 to 2.9.

Thus, for a given amount of fill, this type of PAU will provide a specific and repeatable loading at each speed within its operating range and the load will change with speed according to the built-in power characteristic. If a PAU having a power characteristic of 2.8 is "indexed" or filled sufficiently to provide a load of 3 kw (4.1 HPm) at 65 kph, for example, it will provide the following loads at other speeds:

SPEED (kph)	LOAD (kw)	LOAD (HPm)
10	.02	.02
30	.34	.47
50	1.4	2.0
70	3.7	5.0
90	7.5	10.1
110	13.1	17.8

These speeds and loads will be duplicated until the index point is changed by increasing or decreasing the amount of fluid in the PAU.

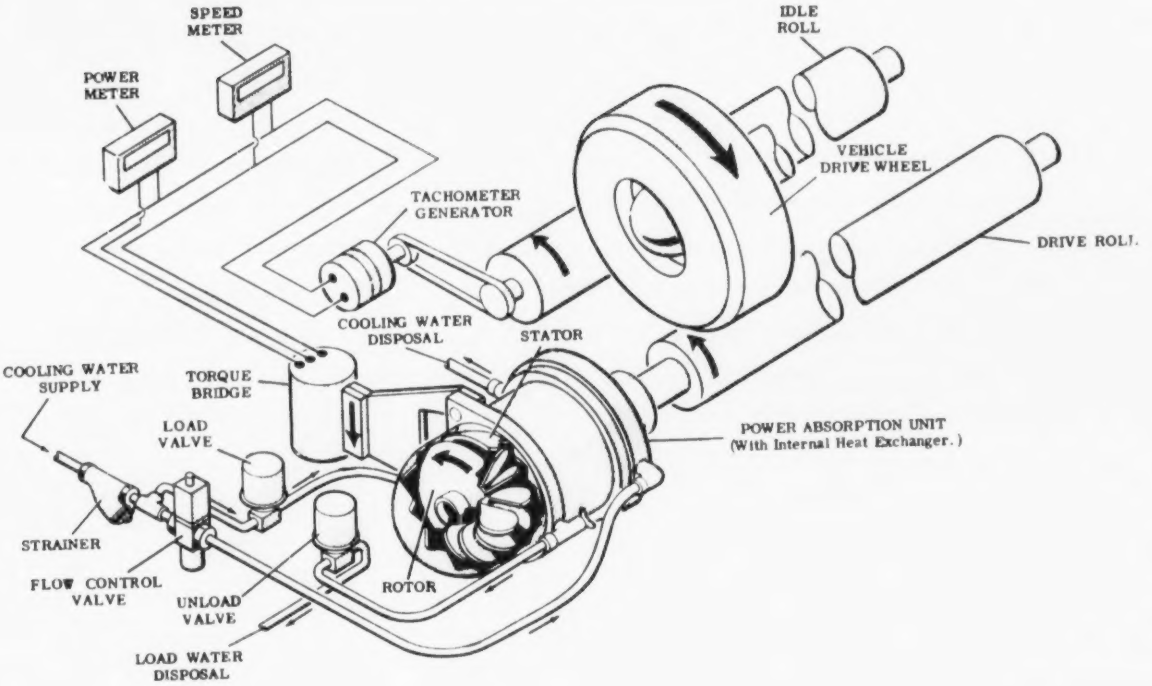

SPEED METER
POWER METER
IDLE ROLL
VEHICLE DRIVE WHEEL
TACHOMETER GENERATOR
DRIVE ROLL
COOLING WATER DISPOSAL
STATOR
COOLING WATER SUPPLY
TORQUE BRIDGE
LOAD VALVE
POWER ABSORPTION UNIT (With Internal Heat Exchanger.)
STRAINER
FLOW CONTROL VALVE
ROTOR
UNLOAD VALVE
LOAD WATER DISPOSAL

Figure 2-9. Principles of Dynamometer Operation

The loads exhibited by a hydrokinetic - closed flow unit having an exponential characteristic of 2.83 has been compared in Table 2-4 with the loads calculated from the EPA generated algorithm. As shown in this table, the errors in PAU load in the lower speed range are small in horsepower, but large in percentage of the calculated PAU load. The influence of such load errors in emissions and fuel economy measurements are unknown.

It is noted that a study by Honda Motor Company, Ltd., which was submitted to the EPA on March 29, 1974, found good correlation between road-loads derived from coast-down tests and chassis dynamometer loads performed with a fixed exponent type of hydrokinetic - closed flow PAU. These results tend to indicate that a more comprehensive study of tire-to-roll losses might show that sufficiently accurate duplication of road-loads for the EPA tests could be obtained without the use of algorithm type of controller.

It is not feasible to use hydrokinetic - closed flow type PAU's for inertia simulation because they lack motoring capability.

Clayton Manufacturing Company furnished information on their Part No. D-18773 Power Absorption Unit. Its rotor inertia is 2.221×10^{-2} kg.m^2, and the cooling water flow required is 2.09×10^{-4} m^3/skw. Price of the basic unit is $2,145, less controls.

The unique capability of the hydrokinetic - closed flow type of PAU to "lock up" a load index point on its built-in exponential characteristic curve is advantageous when it is desired to operate on a fixed curve of speed versus load. However, it would be necessary to add and withdraw fluid from the amount initially locked up in the circulating loop to index the basic load in order to correct for the constant and linear terms in the load algorithm furnished by the EPA. This technique has been used in hydrokinetic PAU's for testing aircraft turboprop engines.

TABLE 2-4

HARLEY-DAVIDSON - 1200 CC

KPH	HORSEPOWER NET PAU	HORSEPOWER PAU LOAD*	HORSEPOWER ERROR	% ERROR
	HP -EPA	2.83 EXP		
20	.26	.12	- .14	54
40	.97	.87	- .10	10
50	1.69	1.63	- .06	4
65	3.43	3.43	0	0
85	7.26	7.33	+ .07	1
110	15.27	15.20	- .07	0.4

HONDA - 90 CC**

20	- .04	.10	+ .14	350
40	.58	.72	+ .14	24
50	1.26	1.35	+ .09	7
65	2.84	2.84	0	0
85	6.31	6.07	- .24	4
110	13.43	12.59	- .84	6

* PAU load indexed @ 65 kph.

** Used EPA data for motorcycle of similar mass to predict Honda 90 road load.

It is possible to re-establish the initial index load at any time because the fluid withdrawn is held in a hydraulic cylinder from which it can be returned to the circulating loop. If, as seems now to be the case, the exponent of the net PAU load versus speed (motorcycle load minus tire/roller loss) is confirmed to vary from 2.4 to 3.1 as shown in Figures 2-2 and 2-3, the above described means of changing the PAU fill could be used to change the fixed exponential characteristic of a hydrokinetic - closed flow PAU as required to improve the accuracy of approximating the road-load curves of individual motorcycles. This technique has been recently demonstrated by Clayton Manufacturing. However, this technique requires substantially more development before it is fully automated. This possibility is suggested because the controller would be less complicated than that needed for algorithm-type loading, and may be substantially less expensive than comparable eddy current and DC motor generators.

2.7.6 Eddy Current

The eddy current PAU consists of a toothed rotor, made of magnetically permeable material such as steel, which rotates in close proximity of magnetized field poles. The poles are arranged so that the lines of flux pass through the teeth of the rotor producing a higher flux density in the electromagnet pole face areas which are in proximity with the rotor teeth and lower flux density in areas between the teeth. As the rotor turns, any given point on the face of each electromagnet pole experiences changes in flux density, resulting in the generation of eddy currents and local magnetic fields which react with those in the rotor teeth. The interreaction of these magnetic fields produces resistance to relative motion between the rotor and stator or torque.

The magnitude of this torque is directly related to the flux density which, in turn, is a function of the current in the coils of the field. The eddy currents in the pole faces generate heat which is dissipated either by passing cooling water within the pole faces or by injecting water directly into the gap between the rotor and stator. The former technique is usually employed in eddy current PAU's of smaller capacity in which the additional hydrodynamic drag is objectionable. Water-in-gap construction is usually preferred in large capacity PAU's because the added drag is acceptable and it permits a more compact size.

The eddy current PAU marketed by Lear Siegler differs from the above in that it is air-cooled and eddy currents are generated in disc rotors at each end of an axial electromagnet assembly. These rotors are finned or slotted to facilitate dissipation of heat.

The basic speed/load characteristics of the eddy current PAU are not acceptable for the motorcycle dynamometer facility because the torque produced at a constant field excitation current tends to be relatively constant. However, highly developed, solid-state closed loop controls capable of following an externally generated electrical signal representing the polynomial speed/load algorithm are available.

In addition to the absorption-only eddy current PAU's, there are universal models available in which an AC induction motor and eddy current clutch are packaged together with the eddy current absorber. These universal models provide motoring capability and could be used for vehicle inertia simulation.

The Dynamatic Division of Eaton Corporation has provided information on their Model 758 DG PAU, rated at 36.8 kw (50 HPm), which is for absorption only. Catalog price of the basic mechanical unit is $6,560. The standard package which also includes controller, torque measuring system, digital tachometer, calibration arms, etc., is priced at $12,260. Their universal Model A15U, which has about the same absorption capacity, plus 11.0 kw (15 HPm) motoring is priced at $23,935 for the basic machine. This would be required for PAU inertia simulation.

Meidensha Electric Manufacturing Company, Ltd. has proposed their Model TW-55 for motorcycle testing. The moment of inertia of the rotor is 4.7×10^{-1} kg.m^2. Cooling water required is given as 1.00×10^{-4} m/^{3}s. The price of this PAU, including solid-state power supply and controller is $15,000.

Lear Siegler, Inc., Industrial Electrical Products Division, also provided literature. Their power absorber is of the eddy current type, but differs in that discs, in which eddy current heating takes place, are air cooled. The windage loss, although probably small at the operational speeds contemplated, is not measured. Their Model C-40 seems to have the necessary capacity for the motorcycle dynamometer facility. The rotor inertia is 3.531×10^{-1} kg.m^2. Price of the basic absorber is $2,100. Although Lear Siegler offers a control for this absorber, it is unsuitable for the motorcycle PAU application because it provides only manual excitation control or automatic constant speed control. A more complex control having an output of 10 amperes at 90 volts, DC would be required and is available from Lebow Associates.

Lear Siegler does not offer a PAU with combined motoring and absorbing capability.

2.7.7 DC Motor/Generator

A DC-type PAU connected to the roller would provide means of not only absorbing power, but also driving the roller as required for simulating vehicle inertia without the use of flywheels. Generally, DC-type PAU systems, when operating in power absorbing modes, are capable of faster response to program inputs and more precise speed regulation than eddy current PAU's. However, they typically are much higher in cost, have an inherently greater moment of inertia, require more complex controls and are more expensive to maintain than universal-type eddy current PAU's.

Meidensha Electric Manufacturing Company, Ltd. submitted information on three DC-type PAU's. All are intended to be used for motorcycle dynamometer systems in which part of the vehicle weight simulation is performed by means of the PAU and controller. The DC-type PAU's range in price from $24,200 to $26,850, including the power supply, controller and some ancillary equipment. Meidensha submitted detailed price information for chassis emission dynamometers and durability units. These are included as Tables 2-5 and 2-6.

Reliance Electric Company suggested their 14.9 kw (20.3 HPm) DC PAU, Type No. 2810, and Controller Type No. S-3R. The approximate price of these components is $20,000. Delivery is currently 35 to 40 weeks.

The Reliance PAU is not trunnion mounted. Torque measurement would require the addition of a brushless-type torque sensor or a torque table which would add $1,500 to $3,000 in cost.

2.7.8 AC Adjustable-Speed Drives

AC induction motors can be operated over a wide speed range by connecting them to a source of variable frequency power. The resulting drive has the advantages when compared to DC drives of lower maintenance due to the elimination of brushes and commutator and speed control without tachometer feedback.

Recent developments in solid-state devices have made such drives more practical in the following way. The basic system for variable frequency AC power supply consists

Table 2-5. PRICE AND SPECIFICATION FOR EMISSION DYNAMOMETER

	ROLLER	FLYWHEEL	DYNAMOMETER WITH CONTROLLER
DC Chassis Dynamometer with Flywheel only	Steel made, 530.5mm in diameter - (ONE)	10 kg + 20 kg + 40 kg + 80 kg + 150 kg direct control	DC Cradled Dynaometer with Controller 15 hp absorption, 11 hp motoring (1,000 rpm-100 km/h)
Total Price: $48,150	$3,650	$20,300	$24,200
DC Chassis Dynamometer with two flywheels and 70 kg Electrical Inertia Simulation	"	80 kg + 150 kg direct control	"
Total Price: $38,950	$3,650	$10,000	$25,300
DC Chassis Dynamometer with one Flywheel and 150 kg Electrical Inertia Simulation	"	150 kg remote control	DC Cradled Dynamometer with Controller 17 hp asborption, 13 hp motoring 1,000 rpm (100 km/h)
Total Price: $33,950	$3,650	$4,600	$25,700
DC Chassis Dynamometer with 300 kg Electrical Inertia Simulation	"	Fixed Flywheel 180 kg including roller and dynamometer inertia mounted on roller shaft	DC Cradled Dynamometer with Control 20 hp absorption, 15 hp motoring 100 rpm (100 km/h)
Total Price: $31,270	$3,650	$770	$26,850
Eddy Current Chasiss Dynamometer with Flywheel only	"	10 kg + 20 kg + 40 kg + 80 kg + 50 kg direct control	15 hp absorption 1,000 rpm model TW-55 cooling water 6L/min.
Total Price: $38,950	$3,650	$20,300	$15,000

Prices of Roller and Flywheel are for your information.

Above price is based on F.O.B. at Yokohama and included boxing charge.

Table 2-6. PRICE AND SPECIFICATION FOR DURABILITY DYNAMOMETER

	ROLLER	FLYWHEEL	DYNAMOMETER WITH CONTROLLER
DC Chassis Dynamometer with Flywheel only	Steel made, 530.5mm in diameter	10 kg + 20 kg + 40 kg + 80 kg + 150 kg direct control	DC Cradled Dynaometer with Controller 25 hp absorption, 19 hp motoring (1,200 rpm-120 km/h)
Total Price: $50,050	$3,650	$20,300	$26,100
DC Chassis Dynamometer with two flywheels and 70 kg Electrical Inertia Simulation	"	80 kg + 150 kg direct control	"
Total Price: $40,500	$3,650	$10,000	$26,850
DC Chassis Dynamometer with one Flywheel and 150 kg Electrical Inertia Simulation	"	150 kg remote control	DC Cradled Dynamometer with Controller 45 hp asborption, 35 hp motoring 1,200 rpm (120 km/h)
Total Price: $40,150	$3,650	$4,600	$31,900
DC Chassis Dynamometer with 300 kg Electrical Inertia Simulation	"	180 kg including roller and dynameter inertia	DC Cradled Dynamometer with Controller 50 hp absorption, 37 hp motoring 750/1,200 rpm (75/120 km/h)
Total Price: $42,820	$3,650	$770	$38,400
Eddy Current Dynamometer with Flywheel only	"	10 kg + 20 kg + 40 kg + 80 kg + 50 kg direct control	Eddy Current Cradled Dynamometer with Controller 25 hp absorption, 1,200 rpm MODEL TW-55 cooling water 9L/min.
Total Price: $38,950	$3,650	$20,300	$15,000

Prices of Roller and Flywheel are for your information.

Above price is based on F.O.B. at Yokohama and included boxing charge.

of converting the 60-cycle power to DC and then converting the DC back to AC by switching or "chopping" at the desired frequency. Early variable-frequency systems were limited by the difficulty of turning off the conduction in silicon-controlled retifiers (SCR's) which were then the most practical high power solid-state switching devices. The recent development of a duplex transistor in a single chip, called the Darlington, has eliminated the complex circuitry which was required to turn off the SCR's. Darlingtons are capable of higher frequency operation so that adverse heating effect of square wave operation of motors can be reduced by six-step square wave pulse-width modulation.

It appears that adjustable-speed AC drives are advantageous in applications wherein sparkless operation, ultra-high speed, multiple-drive synchronization and/or adverse environmental conditions are a factor. None of these seem to be considerations in the selection of a PAU for the motorcycle test facility.

2.8 POWER ABSORPTION SYSTEMS - RANKINGS

2.8.1 Ranking Alternatives - Flywheel Inertia Simulation

Optimum Selection

1. Eddy Current-type PAU, absorption only, control for polynomial speed/load algorithm.
2. Hydrokinetic - closed flow-type PAU, absorption only, fluid displacer and control to correct basic exponential speed/load characteristic to polynomial speed/load algorithm.

Acceptable Selection

3. Hydrokinetic - open flow-type PAU, absorption only, control for polynomial speed/load algorithm.

4. DC motor/generator.

Unacceptable Selection

5. Air blower.
6. Hydraulic pump.

2.8.2 Ranking Alternatives - Partial PAU Simulation Of Inertia

Optimum Selection

1. DC motor/generator-type PAU, motor and absorb, control for polynomial speed/load algorithm plus inertia simulation.

2. Hydraulic pump/motor-type PAU, control for polynomial speed/load algorithm plus inertia simulation.

Acceptable Selection

3. Universal eddy current drive, motor and absorb, control for polynomial speed/load algorithm plus inertia simulation.

Not Appliable

4. Hydrokinetic - closed flow-type PAU.
5. Hydrokinetic - open flow-type PAU.
6. Air blower.

2.8.3 Discussion

Final selection of two, or possibly three, of the alternative systems should be made based upon the following considerations:

- Accuracy required in duplication of motorcycle road-loads to assure equitable emissions and fuel consumption measurements.
- Evaluation of proven, available control systems, versus the probably time and cost for development of more elaborate systems.
- Probability of changing test procedures and consequent adaptability required to avoid obsolescence of equipment.

These selections should then be restudied, obtaining detailed specifications, prices and deliveries from vendors from which to make final choices.

2.8.3.1 PAU's with Flywheel Inertia Simulation

Optimum

The study to date, of available information, tends to indicate that an eddy current- type PAU in combination with a mechanical flywheel weight simulation system and a stock, high performance-type industrial drive controller, plus a computer to generate the control command from the speed/load algorithm is the optimum system for accurate duplication of a wide range of motorcycle performance characteristics, with a minimum of development time and cost.

The hydrokinetic - closed flow-type PAU, with a fixed exponential speed/load curve, in combination with tire-to-roller interaction would be the simplest and lowest cost system. Differences in actual road operational loads, versus the dynamometer loads, need to be evaluated in terms of their influence on emissions and fuel economy measurements to determine if they are significant compared to other uncertainties such as the road data, etc. It is possible to improve the accuracy of the hydrokinetic - closed flow-type PAU's speed/load fit by means of a fluid displacer to correct the basic exponential curve of a hydrokinetic - closed flow unit to the polynomial algorithm by changing the volume of fluid in the working loop. However, a detailed analysis, based upon precise definitions of accuracy and response time requirements, would be necessary to determine the feasibility of this solution.

Acceptable

The hydrokinetic - open flow-type PAU would be acceptable for either fixed exponential or polynomial speed/load testing. However, because the open flow system requires matching of input to output flow, a moderately expensive control system would be required, even for the simpler exponential loading. The control would have to monitor speed and torque and continuously adjust the inlet and outlet flow to maintain the required fill. This would result in a cost disadvantage compared to a hydrokinetic - closed flow PAU system for equal performance.

The addition of a polynomial command signal would provide the ability to more accurately duplicate the road-load, although it is doubtful if the rate of response could be made equal to that of the eddy current system. This system would require engineering development costs greater than those for the eddy current system.

The DC motor/generator, while acceptable, is not optimum because of higher cost. The motoring capability and the faster response to control command signals may be useful with a flywheel inertia simulation system, and might warrant use of this higher cost system.

Unacceptable

The air blower is unacceptable because of its size, its inertia and the complex damper system needed to match loads and power curves.

The hydraulic pump PAU could be used for the polynomial loading requirement. However, the several disadvantages discussed in the test render it unacceptable. The system would require a reservoir, close temperature control heat exchanger to minimize viscosity changes, and a flow control system to produce an approximately cubic speed/load characteristic. Maintenance of filters, etc., would be greater than for a PAU using water as the working fluid.

2.8.3.2 PAU's with Inertia Simulation

Optimum

The DC motor/generator is believed to be the best choice for a dynamometer system in which vehicle inertia is simulated by one to four flywheels plus programmed motoring and braking by the PAU to "trim" the inertia to the 10 kg increments of vehicle mass specified in the NPRM. (This technique is discussed in the Task 4 report.) Primary reasons for ranking the DC motor/generator first are that these machines and controls have been used in complex industrial drives successfully. They would require a minimum of development for adaptation to the motorcycle test facility requirement.

The hydraulic pump/motor and control system has also been used in industrial drives, although, perhaps, not as extensively as the DC motor generator. It is possible that a more detailed dynamometer facility performance specification would result in ranking the hydraulic pump/motor above the DC motor/generator on the basis of fast response time. However, development and maintenance costs would be higher.

Acceptable

Universal eddy current drives are ranked below DC motor/generators and hydraulic pump/motor systems primarily because of their slower response. The acceptability of a slower response to the load command signal depends, ultimately, on how much emissions and fuel economy measurements are affected.

Not Applicable

The hydrokinetic open and closed flow and air blower types of PAU's are not applicable because they lack motoring ability.

Section 3

TASK 3 - EVALUATION OF ROLL CONFIGURATIONS

3.1 INTRODUCTION

This task report will present an analysis of the drive roll assembly for use in the motorcycle chassis dynamometer facility. Performance requirements defined in the EPA Notice of Proposed Rulemaking (NPRM) and Contract No. 68-03-3241 will be examined, and a general specification for the drive roll assembly will be presented. A detailed analysis of the effects of various roller design parameters is presented, and recommendations are made in Section 3.8. In Section 3.3 experiments to quantify the tire/roll interactions will be described and quantitative data will be presented and used to support the design decisions.

3.2 REVIEW OF REQUIREMENTS

The NPRM defines the driving cycle for emissions measurement and durability testing. Both procedures require vehicle operation at idle, steady-state, acceleration and deceleration modes. The maximum speed in the emissions procedure is 90.9 kph; 110 kph (lap 11) in the durability cycle. The power to be transmitted from the motorcycle tire through the roller to the power absorption unit (PAU) has been examined in Section 2. Equations and empirical load factors for several motorcycles were developed in Table 2-1.

This data has been used to select speed/power points for tire to roller tests as discussed in Section 3.3 of this report.

The contract indicates that cradle rolls of 0.203 to 0.229 m (8 to 9 in.) diameter and spaced 0.457 m (18 in.) apart, which have been used in automobile chassis dynamometers, cause undesirable tire flexing, and the use of a single roller of larger diameter is recommended by the EPA for motorcycle testing. This report will quantitatively examine these alternatives. Consideration of the effects of surface texture and crowning of the roller is also included in the design study.

The contract emphasizes the importance of minimum friction and drag in the roller assembly. This is particularly important when testing smaller motorcycles which produce low power outputs. Also, the characteristics of the dynamometer should not be affected by temperature fluctuations and time.

3.3 QUANTIFICATION OF TIRE/ROLL LOSSES

Although the EPA contract indicates that a large diameter, single roller is desirable to minimize tire flexing, little data is available quantifying tire-to-roller interaction for motorcycles. Since these losses plus the PAU and inertia loads constitute the total load on the motorcycle being tested, it was essential that they be measured. An experiment was, therefore, designed to obtain these data as a function of roller size, roll arrangement (single roller or cradle rollers) and motorcycle weight.

3.3.1 Test Description

Three roller diameters, 0.217 m (8.65 in.), 0.324 m (12.75 in.), and 0.508 m (20.0 in.) were fabricated and tested. The two smallest diameters were tested in both cradle and single roller configurations. Only single roller tests were made with the largest roller. The roll centerline spacing was 0.395 m (15.56 in.) for the 0.217 m (8.65 in.) diameter rolls and 0.449 m (17.69 in.) for the 0.324 m (12.75 in.) rolls, which resulted in approximately equal loadings on each of the rolls.

Two motorcycles were used. The larger one, a Harley Davidson FLH, was equipped with a 5.10-16 size tire inflated to 179 KPa (26 psig) pressure cold. The rear axle load was 268.5 kg (592 lbs.). The smaller motorcycle, a Honda 90cc unit, was equipped with a 2.75-17 size tire inflated to 96.5 KPa (14 psig) and 138 KPa (20 psig) cold. The rear axle load was 107.5 kg (237 lbs.). Initial tests were made on the smaller motorcyle with the tire inflated to 96.5 KPa (14 psig), and excessive tire deflection was observed operating on a single 0.217 m (8.65 in.) diameter roller. A second complete test series was run using 138 KPa (20 psig) tire pressure. No tests were performed on the larger motorcycle using reduced tire pressure.

The rollers were carried on antifriction bearings without seals or shields and lubricated with automatic transmission fluid to minimize friction. Adjustable loading was provided by a manually-indexed hydrokinetic-type PAU to ensure stability and for convenience in operation. This type of PAU has an inherent exponential speed/load characteristic; whereby, absorbed power changes approximately as the 2.8 power of rpm. The initial set point or index is determined by the amount of fluid "locked up" in the unit. The speed/load curve remains constant until the amount of fluid in the recirculating loop is changed. Two loading indices for each of the motorcycles were chosen. This selection was based upon an approximation of the road-load predicted by the EPA equation and a higher load representing added power for acceleration. The load index is identified in the plots. The higher letter is used to denote a higher load index; i.e., load "E" is greater than load "D".

Motorcycle rear wheel and PAU torques were measured by means of strain gauge load cells. Motorcycle wheel rpm was measured by an electromagnetic pulse generator and an electronic counter. PAU rpm was measured by means of a DC tachometer generator and a digital voltmeter.

Operating test points were selected to be representative of the speeds and loads appropriate for the EPA emissions test procedure for motorcycles of the two mass classes. Approximately the same speed/load points were run on the following roller setups.

1. Two 0.217 m (8.65 in.) diameter rolls on 0.395 m (15.56 in.) centers.

2. One 0.217 m (8.65 in.) diameter roll.

3. Two 0.324 m (12.75) in.) diameter rolls on 0.449 m (17.69 in.) centers.

4. One 0.324 m (12.75 in.) diameter roll.

5. One 0.508 m (20.0 in.) diameter roll.

3.3.2 Roller Test Results

Laboratory test data are presented in Appendix A and several figures which shall be discussed individually. Torque losses were determined by measuring independently the input torque and PAU, and determining the difference.

Curves through the data points were established using logarithmic curve fit techniques.

It was confirmed that losses in transmitting torque from the rear wheel to the roller become smaller as the roller diameter is increased. This is illustrated in Figure 3-1, a plot of single roller tests on the Honda 90 motorcycle. In this plot, data is provided for two roller diameters and two load indices; the higher indicated by the suffix letter "G".

It can be observed that the torque loss change for a given change in input torque is much smaller with a 508 mm (20 in.) roller than it is with a 200 mm (8.65 in.) roller. For example, increasing the input torque from 20.3 nt m to 47.5 nt m (15.0 ft. lbs. to 35.0 ft. lbs.) on the 20.00 G load curve results in the torque loss increasing from 7.5 nt m (5.5 ft. lbs.) to 8.2 nt m (6.0 ft. lbs.), or a change of 0.7 nt m (0.5 ft. lbs). The same change on the 8.65 G roller results in the torque loss increasing from 15.2 nt m to 34.2 nt m (11.2 to 25.2 ft. lbs.), a change about 32 times as large.

The difference in slope of the "G" and "B" curves indicates that the power level influences the torque loss, possibly due to heating of the tire tread and carcass, but investigation of this was not feasible because of time and budget limitations.

Figure 3-2 shows the torque loss relative to motorcycle wheel speed. The torque losses are observed to increase a greater amount for the smaller roller per unit increase in speed than for the larger roller as was expected.

The small difference in slope between the 20.00 B and 20.00 G curves are noteworthy because torque losses as a function of increased power at a given speed are nearly constant. This is in contrast to the data shown in Figure 3-2 for the smaller roll in curves 8.65 B and G. There, data tend to indicate that the larger, 508 mm (20.0 in.), diameter

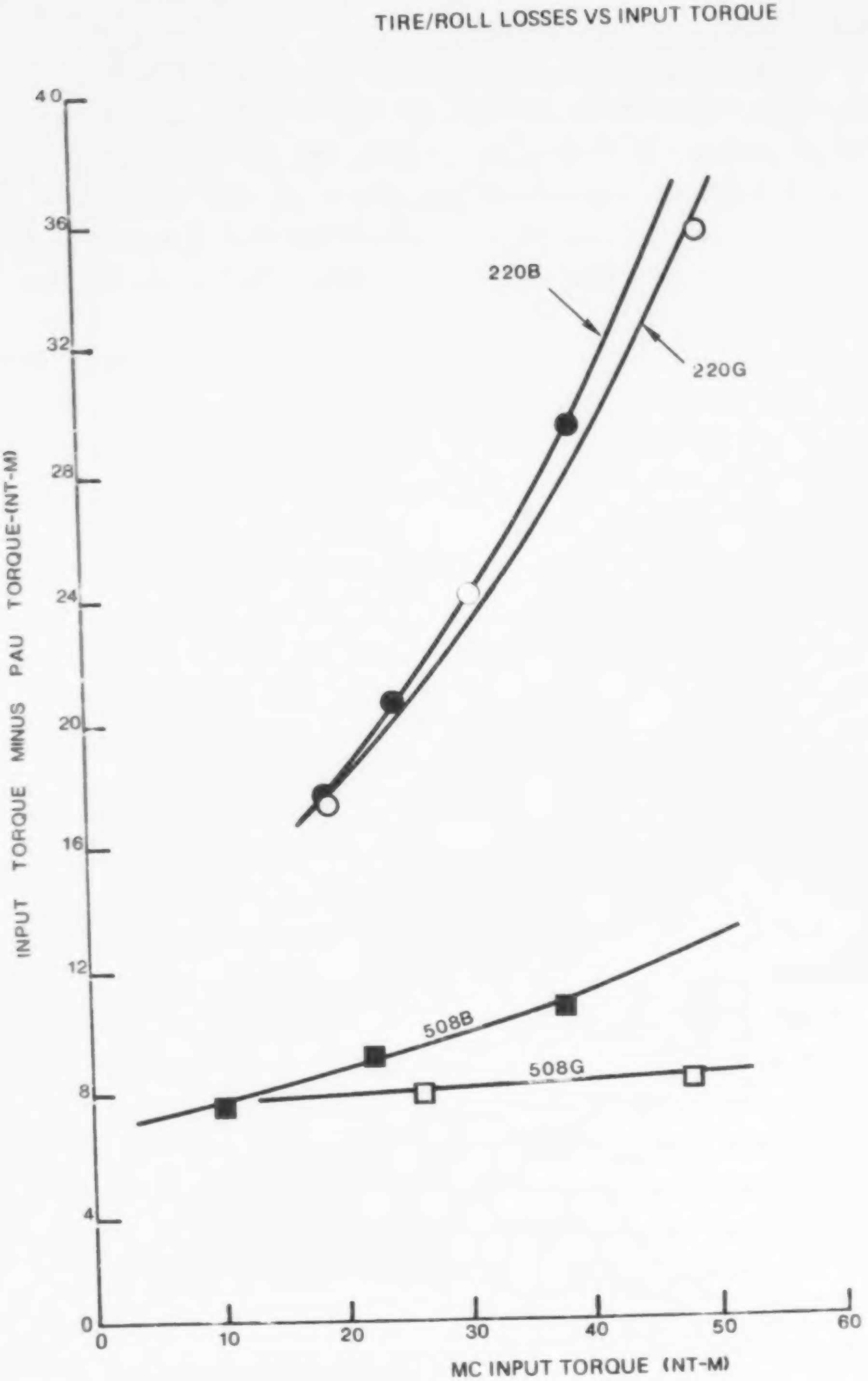


Figure 3-1. Honda Motorcycle Data Single Roll

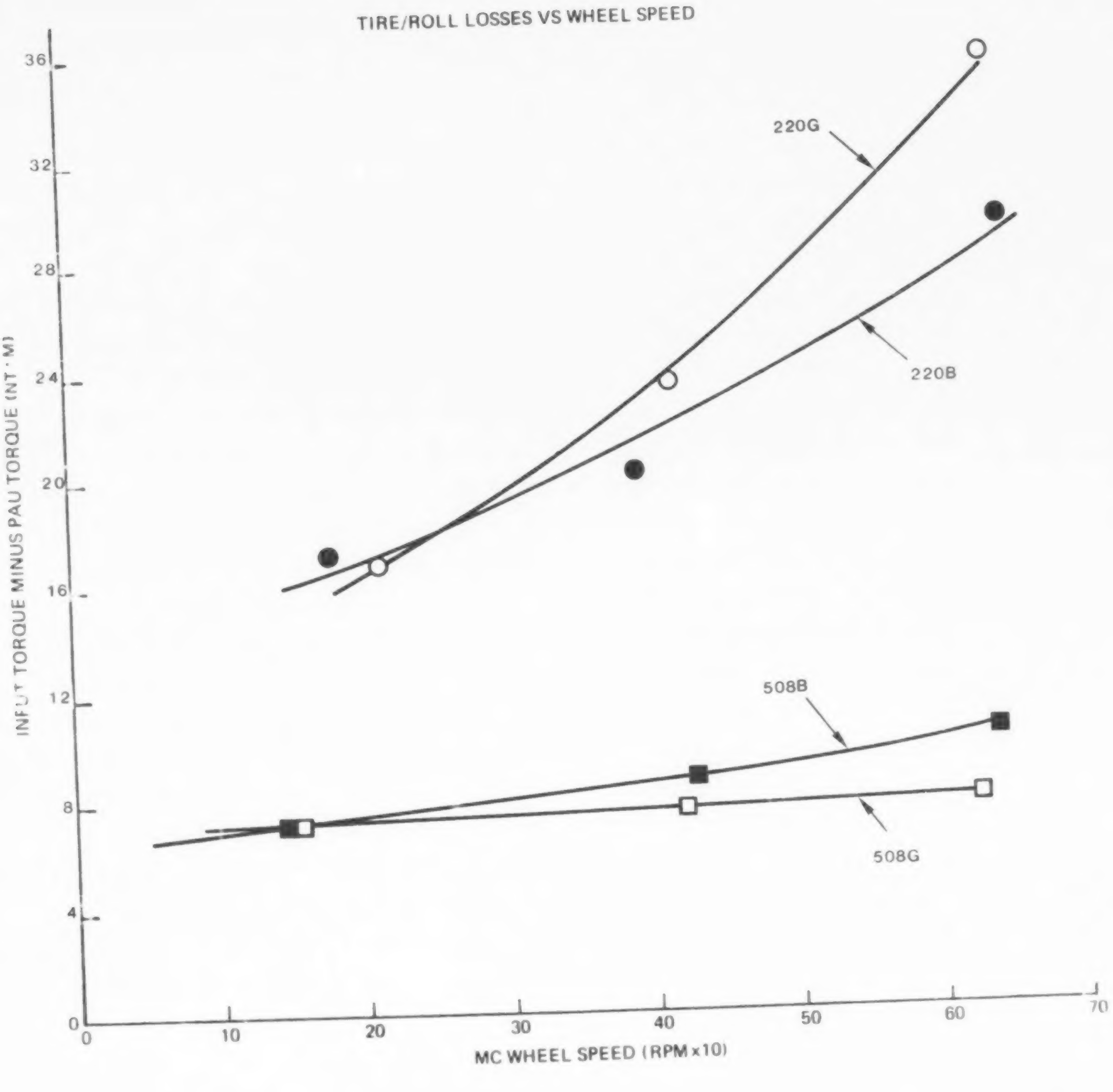


Figure 3-2. Honda Motorcycle Data Single Roll

roller would result in more nearly constant tire/roller losses with changes in load and speed than one of smaller diameter.

Results of operating the Harley Davidson FLH motorcycle on single rolls of 220 mm (8.65 in.) and 324 mm (12.75 in.) diameter are plotted on Figures 3-3 and 3-4 respectively. Again, it is observed that the torque losses are reduced by increasing the roller diameter.

These curves indicate that the change in torque loss per unit change in motorcycle input torque is reduced by increasing the roller diameter. Also, increased motorcycle input torque results in smaller changes in torque loss per unit change in input torque as indicated by the "E" curves and the "D" curves in Figures 3-3 and 3-4. This is thought to be the results of increased tire temperature, although no tire temperature data was recorded to substantiate this.

The relationship between roller diameter, speed and torque loss for the larger motorcycle is plotted on Figure 3-4. The larger roller was again observed to minimize the change in torque loss for a unit change in speed. These data tend to indicate that the 508 mm (20.00 in.) roller is superior to the 220 mm (8.65 in.) or 324 mm (12.75 in.) rollers because it results in minimum torque loss at typical input torques and operating speeds. It is also believed that the 508 mm (20.00 in.) diameter is probably near optimum, if intrinsic inertia, installation space and cost are considered. However, much more extensive experimentation would be necessary to adequately support this opinion. Such experiments should include a large variety of motorcycles and axle loads, additional tire types and sizes and more test points per combination.

The comparative performance of single roll and cradle roll arrangements can be seen in Figures 3-5 and 3-6. As in the data obtained with the single roll, losses decrease as roll size increases. Although only limited data

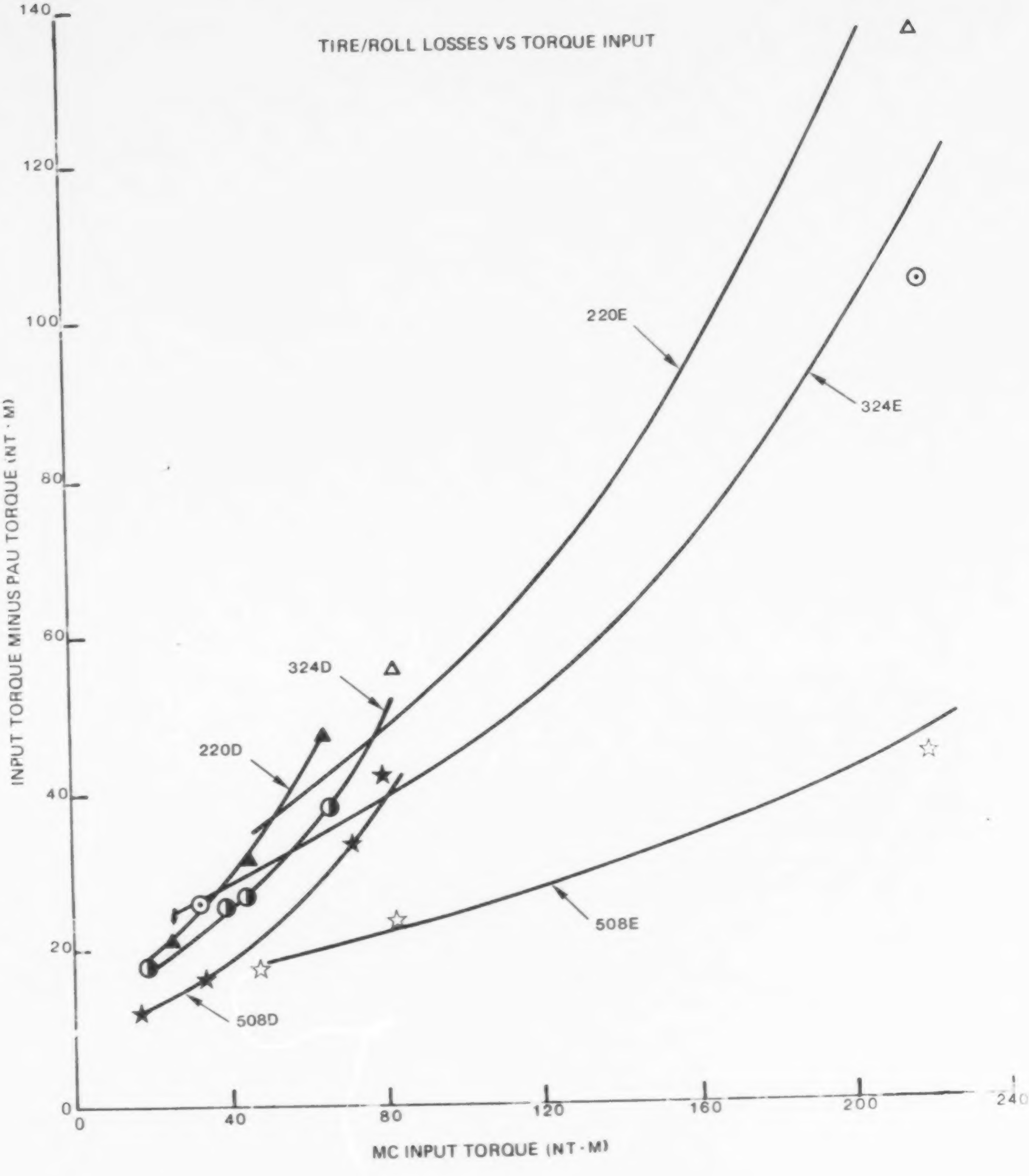


Figure 3-3. Harley Davidson Motorcycle Single Roll

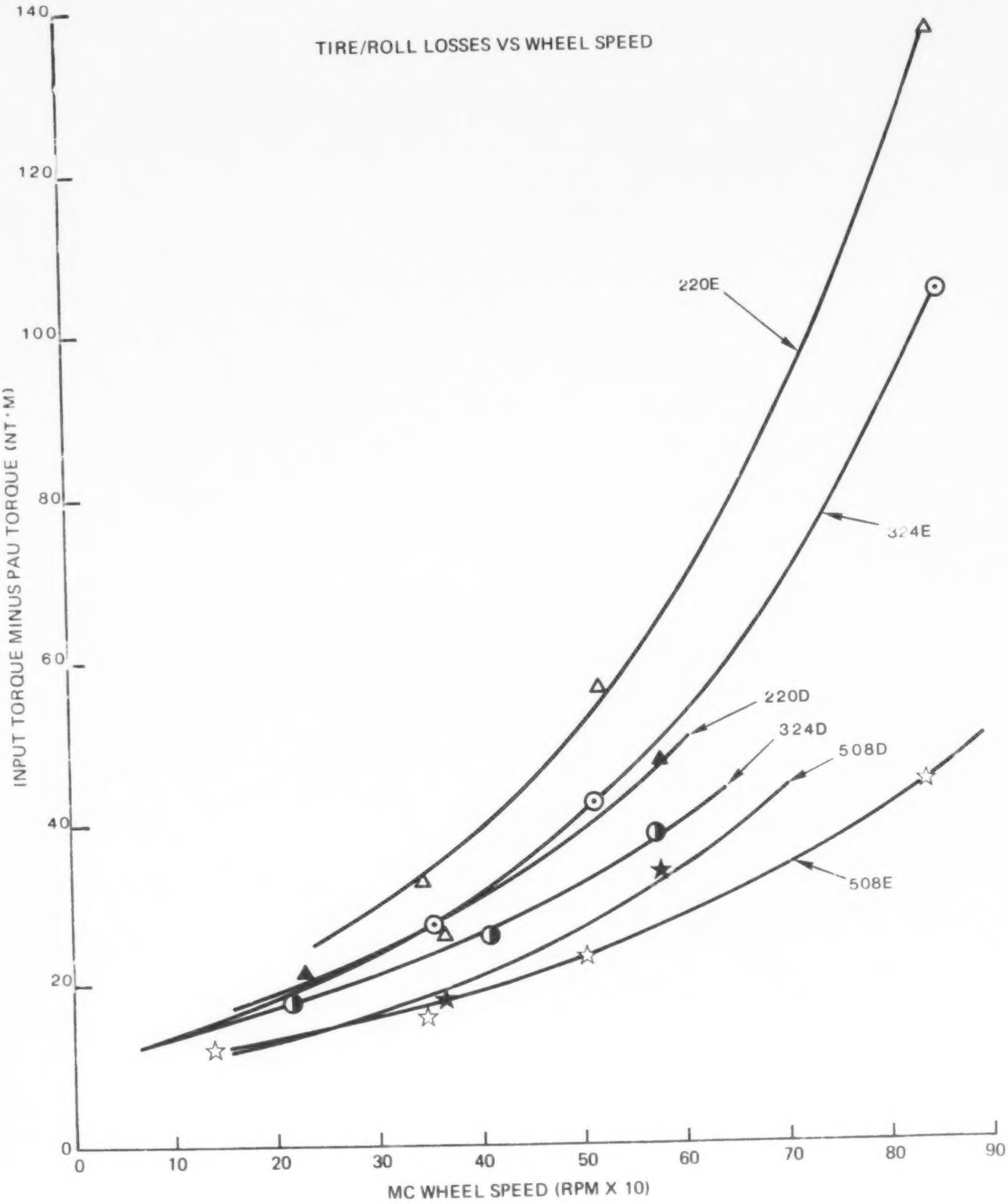


Figure 3-4. Harley Davidson Motorcycle Single Roll

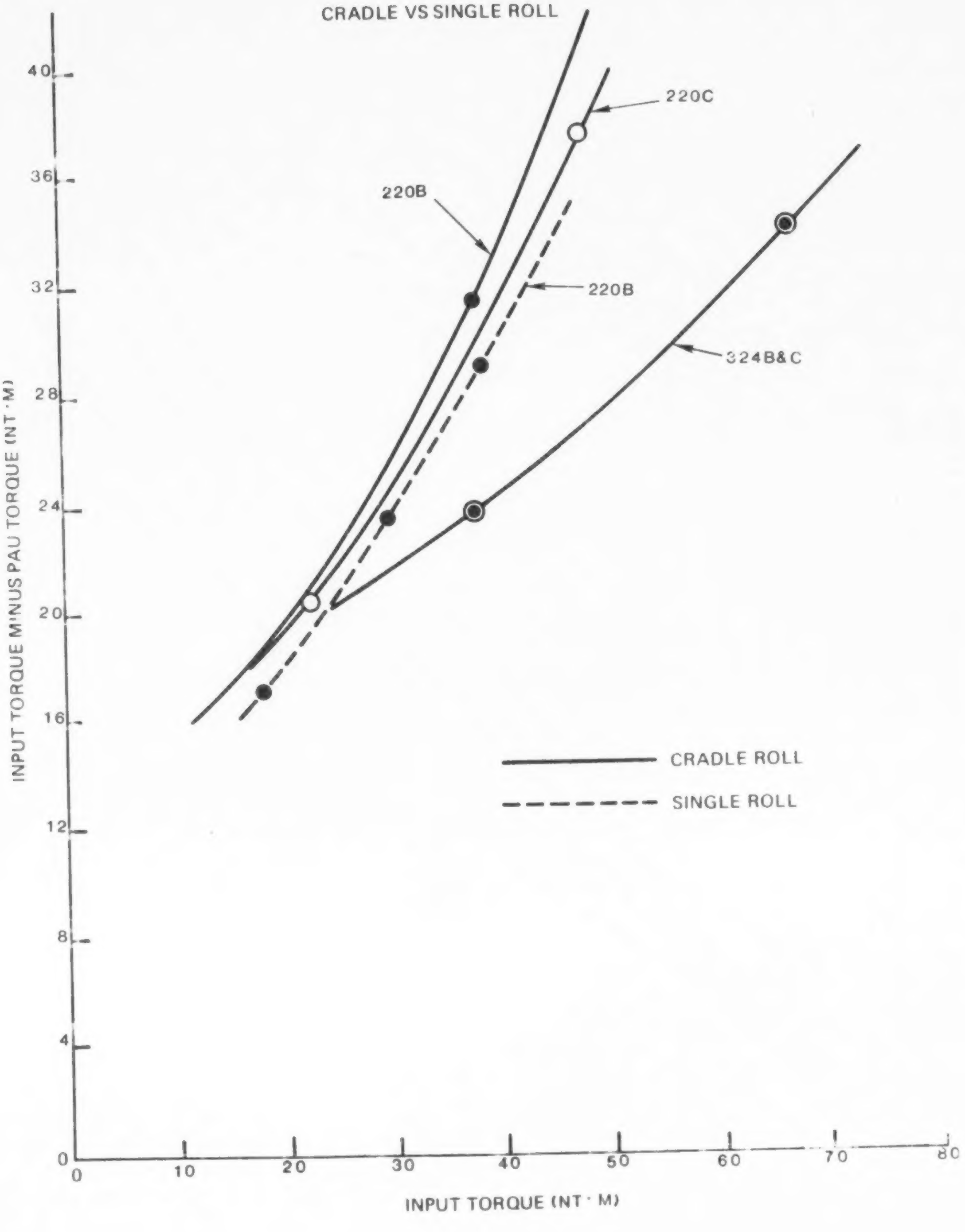


Figure 3-5. Honda Motorcycle Data

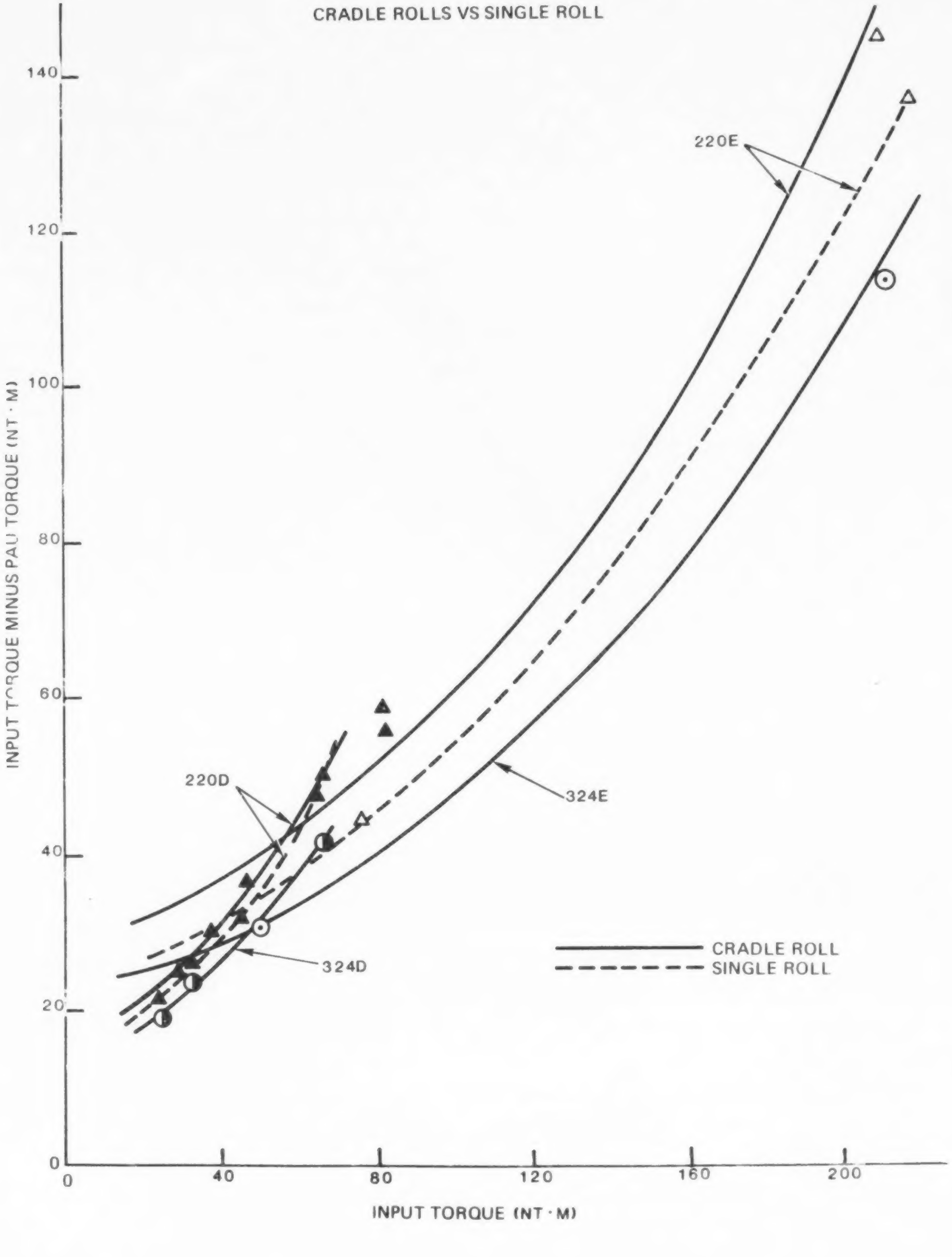


Figure 3-6. Harley Davidson Motorcycle Data

was obtained, it appears that the losses experienced with single rollers are slightly less than those resulting from use of cradle rollers.

As shown in Appendix A, a series of tests were conducted with the Honda bike at two different tire inflation pressures. Data was limited and no discernible relationship between tire/roll losses and tire inflation pressures could be established.

During the course of the test program, utilizing a single roller configuration, it was observed that tire/roll losses increased if the rear wheel of the motorcycle was not properly positioned atop the roller. If the motorcycle's rear axle was positioned either in front or behind the top dead center of the roll the losses appeared to increase.

3.4 ROLLER SURFACE CONTOUR

Making the diameter of the roller larger in the center than at the ends is a means of causing the tire contact patch to be centered automatically on the roller, despite limited skewness between the wheel and roller axes. This construction is used in the Hartzell Mark II motorcycle dynamometer.

The operating principle is sketched in Figure 3-7. A wheel rolling with its axis inclined to the surface is shown in Figure 3-7a. It will describe a circular track with radius 'r' as shown in Figure 3-7b. The radius of its path will be the radius of the wheel axis with respect to the surface. For example, a 610 mm (24 in.) diameter wheel inclined 3 degrees will roll on a circle of approximately 5800 mm (229 in.) radius.

Similarly, a wheel running on a crowned or tapered surface roller will travel in a spiral path toward the larger diameter of the roller as shown in Figure 3-7c.

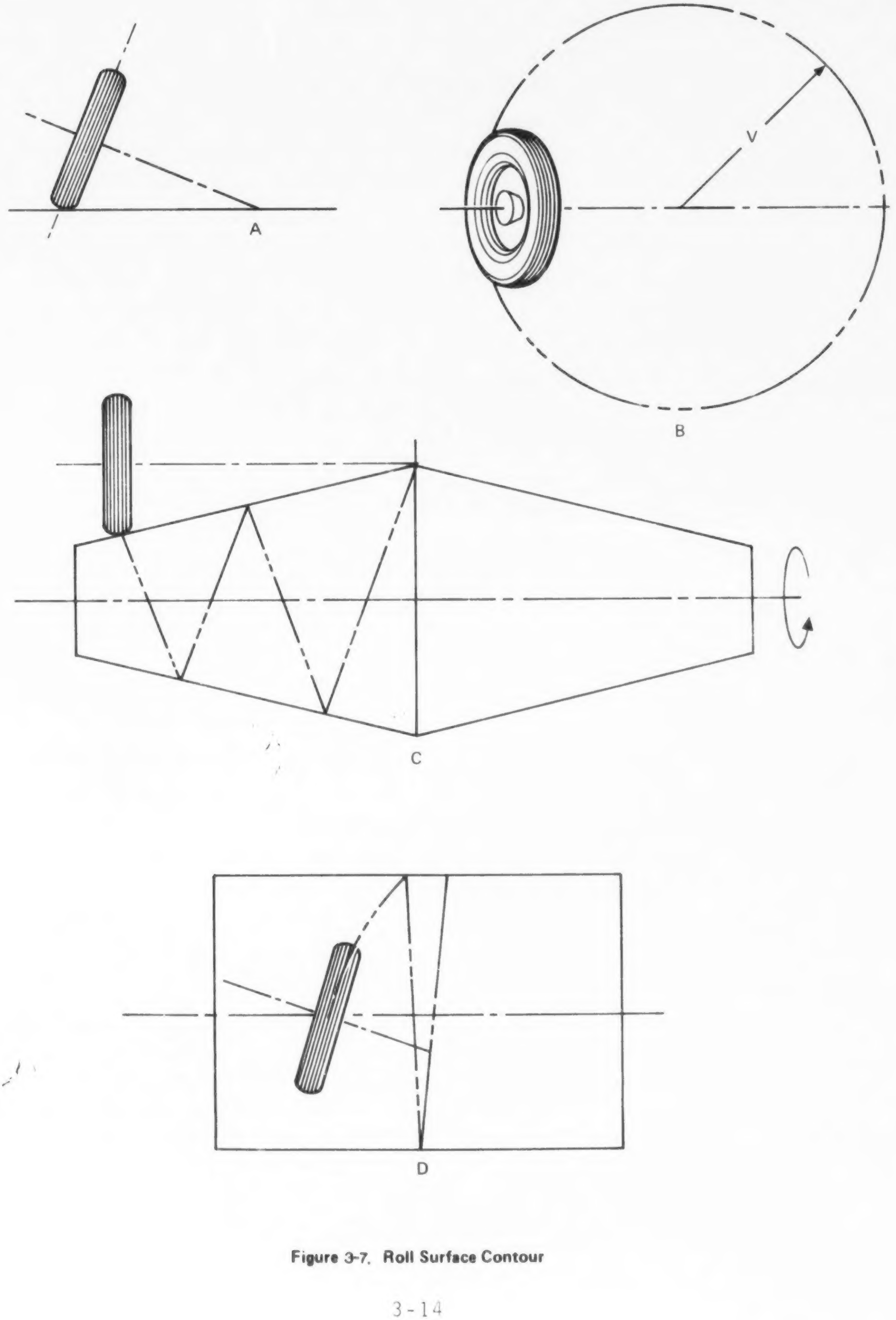


Figure 3-7. Roll Surface Contour

Equilibrium of the forces acting on the tire contact patch will occur at a point near the maximum diameter if, in this position, the wheel and roller axes are parallel. The distribution of contact area will be unequal and increased losses due to tread creep will result if, when the tire patch is approximately centered over the maximum roller diameter, there is skewness between the axes. This loss does not occur on a cylindrical roller. If the skewness between the axes is in excess of that which can be accommodated by the amount of taper or crown, the tire will run off of the roller.

Automatic alignnment of the wheel and roller axes, rather than centering of the tire contact patch, occurs on straight cylindrical rollers as shown in Figure 3-7d. The center of the contact patch describes a helix of decreasing pitch until the axes are aligned. With a headfork angle of typically 25 degree to 30 degree, motorcycles tend to reduce the pitch of the the helix, but no instability has been observed in operation. When the front wheel clamp is not centered or there is frame or wheel axle misalignment, the tire will not run in the center of a straight cylindrical roller, but there is no adverse effect on performance. Similar misalignment could result in the tire operating at a considerable slip angle, although laterally centered, on a crowned roller.

3.5 SURFACE FINISH

Roll surface finish and contamination affect the coefficient of friction and the maximum torque which can be transmitted between the motorcycle tire and roller. Experience from automobile chassis dynamometer operation is probably applicable, even though there are some differences in tire cross section, unit loading, etc.

Application of hard grit coating is of value in preventing skidding or tire breakaway where water, oil or dirt is present. However, there is no appreciable advantage when the tire and roll surface are reasonably clean and dry. The disadvantages of abrasive coating the roller are: increased tire tread wear, which may be hazardous during prolonged tests, and the coating of the roller with tread compound under some circumstances. Another consideration in the use of grit coating is that the surface gradually changes due to dulling of the particle edges and loss of particles due to bonding failure.

Both circumferential and axial grooving have been tried. These grooves also tend to increase tire wear when new and lose effectiveness as the edges of the grooves become rounded. A combination of right and left hand spiral grooves, resulting in an elongated diamond pattern, has recently been developed by Clayton Manufacturing. It seems to be quite effective in maintaining traction even with wet or dusty tires, yet does not show evidence of tread wear or tendency to become coated with rubber.

3.6 MATERIAL OF CONSTRUCTION - ROLLER

Aluminum, with an anodic surface treatment, has been used for roller fabrication. Although it is reputed to have a higher coefficient of friction in combination with tire tread stock than that for steel, it is doubtful if there is a demonstrable, practical improvement in actual operation. There is little to be gained in reduced roller inertia since an emissions test dynamometer must simulate the inertia of the vehicle.

3.7 ROLLER MOUNTING

The roller shaft bearings should be selected to provide a B-10 life of 10,000 hours or more, as defined by the Anti-Friction Bearing Manufacturers Association, to assure reliability and minimize maintenance. The use of low friction seals or shields and oil lubrication would result in low and stable drag losses, which are desirable for the contemplated services.

3.8 RECOMMENDATIONS

Based upon the results of laboratory data and detailed discussion of design features in Section 3 of this report, the following roller specifications are proposed.

Type - single roller
Diameter - 530.5 mm (20.9 in.)
Width - 300 mm (12 in.)
Material - low carbon steel
Surface - cylindrical (uncrowned)
Surface texture- smooth

The limited tests performed are believed adequate to be insure that the roller specified above will operate satisfactorily in the contemplated motorcycle dynamometer.

Section 4

TASK 4 - EVALUATION OF INERTIA SIMULATION METHODS

4.1 INTRODUCTION

This section will present an analysis of methods for the simulation of vehicle inertia with a chasiss dynamometer. The EPA Proposed Rulemaking (NPRM) and Contract No. 68-03-2141 define inertia requirements, and these are examined. A general specification for the inertia simulation system is presented. An analysis of the two basic methods available for inertial simulation is detailed in Section 4.4 of this report and recommendations are made in Section 4.5.

4.2 REVIEW OF REQUIREMENTS

The NPRM defines the driving cycle for emissions measurements in Appendix I and the durability schedule in Appendix IV. Inertia requirements for both cycles are the same, but depending on the simulation technique, they may impose different specifications for meeting the requirements of the two cycles. The contract stipulates that if flywheels are used for inertia simulation, then they shall be coupled to the roller automatically by means of remote control such as electric switches or pushbuttons and that such control shall be fail-safe to prevent injury to the operator or damage to equipment in the event of loss of electrical power, air supply, etc. A further requirement is that if flywheels are driven through a speed increaser, such as a gearbox, then it shall be a low-friction device to avoid inaccuracy of vehicle loading resulting from unmeasured losses in the drive.

Increments of 10 kg from 100 to 700 kg loaded-vehicle mass are specified in the contract with consideration to be given to reducing this maximum from 700 to 500 kg as a means of reducing system complexity and/or improving accuracy or precision. Similarly, the size of the minimum increment influences the system performance and cost. The possibility of using smaller increments such as 5 kg, for smaller vehicles especially, to improve accuracy and the use of larger, possibly 20 kg, increments to reduce complexity, should be evaluated.

It is indicated that both the use of multiple flywheels and/or the use of programmed motoring and absorbing by the power absorption unit (PAU), are to be considered for vehicle inertia simulation.

Neither the contract or the NPRM indicate the permissible tolerance on accuracy or response time for the inertia simulation system. However, emissions will be influenced by the vehicle inertia simulation accuracy, since the horsepower which must be developed at the driving wheel during acceleration is directly affected by the inertia.

4.3 GENERAL SPECIFICATIONS

In light of the requirements for emissions tests as proposed in the NPRM and the contract, a basic set of minimum specifications can be established. These are listed in Table 4-1 and 4-2. Two specifications are called out because of the differences imposed by utilizing mechanical or electrical simulation.

Table 4-1. GENERAL SPECIFICATION FLYWHEEL INERTIA SYSTEM

Range	100 to 700 kg equivalent vehicle weight in 10 kg increments with optional reduction to 500 kg.
Accuracy	Plus or minus 1.5 percent of selected value.
Control	Remote switches or pushbuttons.
Function	Fail safe in case of loss of electric power, air pressure, etc.
Interlock	Range change blocked out above zero rpm in flywheel system.
Response Time	Not applicable to flywheel system.
Windage and Friction Losses	Inertia simulation system not to introduce error greater than 5 percent of road load force at any operating point.

Table 4-2. GENERAL SPECIFICATION ELECTRICAL INERTIAL SIMULATION

Range	100 to 700 kg equivalent vehicle weight in 10 kg increment with optional reduction to 500 kg.
Additional PAU absorption capability	18.4 kw (25 HPm).
PAU motoring capability	18.4 kw (25 HPm).
Accuracy	Plus or minus 1.5 percent selected value at any speed above 5 kph and 5 percent from 1 to 5 kph.
Control	Remote switches of pushbuttons.
Function	Fail safe in case of loss of electric power, air pressure etc.
Interlock	Range change blocked out above zero rpm.
Response Time	0.5 sec. for 90 percent of step change in PAU type simulation system as measured in force change.
Windage and Friction Losses	Inertia simulation system not to introduce error greater than 5 percent of road load at any operating point.

4.4 INERTIA SIMULATION SYSTEM CHARACTERISTICS

In this section, the performance, suitability and probable cost of mechanical-type and PAU-type inertia simulation systems shall be compared as they relate to the requirements of motorcycle emissions testing and durability driving cycles. Design and operation characteristics shall be considered as follows:

- Calibration
- Accuracy of simulation
- Response to change in rpm
- Stability of calibration
- Convenience
- Maintenance

The primary consideration in comparing inertia simulation systems shall be the emissions test procedure as defined in the EPA NPRM and Contract No. 68-03-2141. Durability driving requirements shall be regarded as secondary.

4.4.1 Mechanical Inertia Simulation System

The flywheel is a primary standard of inertia. As proposed for this application, the unit consists of coupling combinations of flywheels to the common shaft of the roller and PAU so that the combined inertia is equivalent to that which is required; i.e., 100 to 700 kg in 10 kg steps. This system would consist of a fixed flywheel which, in combination with the PAU and drive roller inertias, would be equivalent to 100 kg vehicle inertia weight. Additional flywheels of 10, 20, 40, 80, 160 and 320 kg equivalent vehicle inertia would be engaged singly or in combination to provide the 60 increments required.

The mechanical inertia system is constructed in the following manner. A drive shaft, which is coupled to the roller shaft and one fixed inertia wheel, is carried on antifriction bearings. The additional flywheels are carried on antifriction bearings and each can be individually coupled to the shaft by means of a latching-type air-operated clutch. Clutch actuation and selection of the flywheels is controlled from the operator's station by means of electrical switches or pushbuttons. An electrical interlock is provided so that the clutches can neither be engaged or disengaged when the flywheels are in motion.

Maintenance, consisting of lubrication, adjustment of clutches and electropneumatic actuators would not be appreciably more than necessary for the PAU and roller.

The windage, bearing friction and seal drag of the flywheels and roller can approach the total load of a small motorcycle. This unmeasured loss varies with the number of flywheels engaged and with the operating rpm. However, this loss is not expected to change as a function of system usage. A value of 4 Nt.m (3 lbs. ft.) torque, rising slightly with rpm is expected. This loss has been estimated based on the performance of existing dynamometers used in the testing of light-duty vehicles. Data on those systems indicate that system drag characteristics are quite stable after initial break-in. This loss is equivalent to the torque exerted by the rear wheel of a Honda 90cc unit rotating at 20 rpm. Although it can be minimized by careful design, elimination of this variable error is desirable. This can be done by connecting a small (3 kw) motor to the shaft, programmed to apply driving torque to compensate for the windage and frictional losses of the flywheels engaged at the instantaneous speed of operation. Such a compensating motor and control could be used in conjuction with PAU's incapable of motoring, or the PAU absorption characteristics could be programmed to take into account this inherent loss. PAU's having motoring capability could be utilized for compensating flywheel losses by modifying the calculation of road-load and acceleration force as required. It is expected that sufficient accuracy could be obtained with a simple control. The flywheel windage and friction compensation calibration could be checked by driving the roller up to a given rpm and observing whether the rpm increased or decreased during coasting.

Since the flywheel is a primary standard of inertia, it requires no periodic calibration checks unless it is damaged. The response to a change in rpm is instantaneous for all practical considerations. This is important since any perceptible time lag could adversely affect the accuracy of fuel consumption and emissions determinations, particularly in those portions of the test cycle which involve gear changing or braking.

The flywheel inertia system has greater mechanical complexity and requires more space for installation. Maintenance, consisting of lubrication, clutch service, etc., would not add significantly to that required on the PAU and roller.

The use of a speed-increasing device, such as a gearbox, between the low-speed roller shaft and the flywheel system is attractive as a means of reducing the size of the flywheels, bearing and clutches. There are two negative factors to be considered. One is the loss in the transmission of power due to pumping losses in the gears and bearings and due to seal friction, etc. These losses can be sizeable compared to the inertial power-loading appropriate for small motorcycles, and they are difficult to calibrate out of the system or otherwise compensate for because they vary with temperature of the lubricant, gear-loading and length of service. The second negative factor is that the added cost of a gearbox may be greater than that of increasing the size of the inertia simulation system components.

The difference in inertial simulation system complexity, resulting from changing the 10 kg increments to 5 kg or 20 kg increments, is relatively small in terms of mechanical system; or the number of programming selector switch positions for a PAU inertia simulation system would change, resulting in the addition or deletion of one binary step increasing the number of inertia steps to 120 or reducing the number to 30. The change in cost is estimated to be less than $8,000 for mechanical system and much less for a PAU inertia simulation system. However, it is not possible to assess the effect on emissions and fuel economy performance of either change at this time.

4.4.2 PAU Inertia Simulation

Using a PAU having both motoring and driving capability programmed to supplement the intrinsic inertia of the roller, etc., is an attractive method for the emission test cycle inertia simulation because it eliminates the need for a complex flywheel system and provides closer simulation than 10 kg increments. In this system, an on-line calculation of the sum of the forces at the rear wheel due to road-load at the instantaneous velocity, and the instantaneous rate of change in velocity is made. The force is adjusted for the intrinsic inertia of the roller, PAU, etc., and then this net instantaneous force is compared to the observed force (PAU torque) and the appropriate error signal causes the PAU to drive or brake as necessary to make the calculated and measured forces (torque) equal. The instantaneous force equation is:

$$F_T = F_o + F_1 V + F_2 V^2 + m \frac{dV}{dt} \qquad (4\text{-}1)$$

where

F_o, F_1 and F_2 = empirical factors defined by EPA data
V = velocity in meters/second
m = vehicle mass in kg
t = time in seconds

In the PAU system of inertia simulation, a digital or analog computer calculates in real time the roller torque or load which is appropriate for the instantaneous speed and change in speed based upon vehicle mass. The actual torque and the calculated torque are compared and the difference, if any, causes the controller to increase or decrease the PAU torque, as required. The principal difficulty with this method of vehicle inertia simulation is that the response time of the PAU and the roller speed-sensing system result in some lag between the program and load. This is of less consequence in performing the polynominal road-load curve because the changes in PAU loading per unit time are relatively small. However, the inertia simulation part of the control command is determined by the rate of change of speed or derivative which changes rapidly and is small in magnitude.

The roller speed signal can be obtained from a DC tachometer generator or a pulse rate generator. Both exhibit noise in the output signal; pole ripple of typically 0.5 percent (peak to peak) of the signal for the DC tachometer, or 0.05 percent pitch error in the spacing of pulse generator teeth resulting in frequency modulation of the pulse rate. It is assumed that either generator is driven directly from the roller shaft, but often there is additional error introduced in the drive means between the tachometer generator and roller shaft. In either case, filtering of the speed signal must be performed to prevent the noise portion of the tachometer output from partially saturating the control amplifier. This filtering introduces a time constant of, typically, 0.25 to 0.5 seconds.

In addition to the time lag due to filtering, the PAU and controller time lag must be added. Both DC motor/generators and universal eddy current PAU's have considerable inductance, and the rate of change in excitation which can be used is limited by the point at which control instability is reached. The hydraulic pump/motor type of PAU in combination with a sophisticated control is potentially faster in response, but probably more expensive.

An additional concern is a system calibration. A reliable and accurate technique for calibrating or verifying the calibration of this system must be established. At this time, no accurate techniques or procedures are known.

4.4.3 PAU-Type Inertia Simulation with Added Flywheel Inertia

The addition of mechanical inertia to the system would reduce the potential error in PAU inertia simulation experienced by the motorcycle under test, and decrease the maximum PAU power required. A single flywheel could be added to make the total inertia of the dynamometer system equal to the mean of the motorcycles to be tested, but the time lag error in the PAU simulated portion of the total inertia would tend to penalize vehicles of mass lower than the mean and reduce the loads on those greater than the mean. Use of additional, clutchable, flywheels would further lessen the loading errors, but would partially defeat the benefits of using PAU inertia simulation by increasing mechanical complexity.

4.4.4 Costs

It is difficult, at this time, to provide pricing information about the alternative discussed. However, some relative magnitudes and comparisons can be "guesstimated" from quotes provided by Meidensha Electric Mfg., Co., and these have been included in Tables 4-3 and 4-4.

4.5 RECOMMENDATIONS

4.5.1 Ranking Alternatives

Optimum Selection
Multiple Flywheel system

Acceptable Selection
Single fixed flywheel with inertia corrected command signal to motor/absorber type PAU.

Unacceptable Selection
Full electrical simulation utilizing controls which provide command signals to a motor/absorber type PAU.

4.5.2 Discussion

The multiple flywheel system is deemed the optimum selection because of its inherent stability of calibration, and the absence of response time lag. Direct-drive variable inertia systems have been used successfully in automotive emissions testing facilities for some time and result in minimum losses. The use of gearboxes, cog belts or other indirect means of coupling the flywheels to the roll, can result in significant losses. Therefore, unless these losses can be eliminated, indirect couplings should be avoided.

The PAU-type of inertia system, in combination with a single flywheel, is an acceptable selection because

Table 4-3. PRICE AND SPECIFICATION FOR EMISSION DYNAMOMETER

	ROLLER	FLYWHEEL	DYNAMOMETER WITH CONTROLLER
DC Chassis Dynamometer with Flywheel only	Steel made, 530.5mm in diameter - (ONE)	10 kg + 20 kg + 40 kg + 80 kg + 150 kg direct control	DC Cradled Dynaometer with Controller 15 hp absorption, 11 hp motoring (1,000 rpm-100 km/h)
Total Price: $48,150	$3,650	$20,300	$24,200
DC Chassis Dynamometer with two flywheels and 70 kg Electrical Inertia Simulation	"	80 kg + 150 kg direct control	"
Total Price: $38,950	$3,650	$10,000	$25,300
DC Chassis Dynamometer with one Flywheel and 150 kg Electrical Inertia Simulation	"	150 kg remote control	DC Cradled Dynamometer with Controller 17 hp asborption, 13 hp motoring 1,000 rpm (100 km/h)
Total Price: $33,950	$3,650	$4,600	$25,700
DC Chassis Dynamometer with 300 kg Electrical Inertia Simulation	"	Fixed Flywheel 180 kg including roller and dynamometer inertia mounted on roller shaft	DC Cradled Dynamometer with Control 20 hp absorption, 15 hp motoring 100 rpm (100 km/h)
Total Price: $31,270	$3,650	$770	$26,850
Eddy Current Chasiss Dynamometer with Flywheel only	"	10 kg + 20 kg + 40 kg + 80 kg + 50 kg direct control	15 hp absorption 1,000 rpm model TW-55 cooling water 6L/min.
Total Price: $38,950	$3,650	$20,300	$15,000

Prices of Roller and Flywheel are for your information.

Above price is based on F.O.B. at Yokohama and included boxing charge.

Table 4-4. PRICE AND SPECIFICATION FOR DURABILITY DYNAMOMETER

	ROLLER	FLYWHEEL	DYNAMOMETER WITH CONTROLLER
DC Chassis Dynamometer with Flywheel only	Steel made, 530.5mm in diameter	10 kg + 20 kg + 40 kg + 80 kg + 150 kg direct control	DC Cradled Dynaometer with Controller 25 hp absorption, 19 hp motoring (1,200 rpm-120 km/h)
Total Price: $50,050	$3,650	$20,300	$26,100
DC Chassis Dynamometer with two flywheels and 70 kg Electrical Inertia Simulation	"	80 kg + 150 kg direct control	"
Total Price: $40,500	$3,650	$10,000	$26,850
DC Chassis Dynamometer with one Flywheel and 150 kg Electrical Inertia Simulation	"	150 kg remote control	DC Cradled Dynamometer with Controller 45 hp asborption, 35 hp motoring 1,200 rpm (120 km/h)
Total Price: $40,150	$3,650	$4,600	$31,900
DC Chassis Dynamometer with 300 kg Electrical Inertia Simulation	"	180 kg including roller and dynameter inertia	DC Cradled Dynamometer with Controller 50 hp absorption, 37 hp motoring 750/1,200 rpm (75/120 km/h)
Total Price: $42,820	$3,650	$770	$38,400
Eddy Current Dynamometer with Flywheel only	"	10 kg + 20 kg + 40 kg + 80 kg + 50 kg direct control	Eddy Current Cradled Dynamometer with Controller 25 hp absorption, 1,200 rpm MODEL TW-55 cooling water 9L/min.
Total Price: $38,950	$3,650	$20,300	$15,000

Prices of Roller and Flywheel are for your information.

Above price is based on F.O.B. at Yokohama and included boxing charge.

it would be a satisfactory means of inertia simulation if a suitable PAU and control are selected, and it is determined that the response time lag does not reduce the accuracy of emissions, durability and fuel consumption test results to an unacceptable degree. This approach is far superior to the technique which utilizes full electrical inertia simulation (corrected for the inherent inertia of the roll, etc.) because of its improved response time and time lag characteristics.

Another alternative is a system employing multiple flywheels and motoring capability. The flywheels would still functions as the principal method of simulating vehicle inertia. The motoring capability could be used to perform automated coast-down calibrations and as compensation for the inherent losses of the system.

Section 5

TASK 5 - EVALUATION OF VARIABLE FLOW COOLING SYSTEMS

5.1 INTRODUCTION

This task report will present a general analysis of blowers and blower control systems suitable for incorporation in a variable flow blower system to provide cooling air for motorcycle engines. The pertinent regulations which affect the design of this system will be examined, and a general set of specifications will be presented. A review of blower systems currently being utilized in commercially available motorcycle chassis dynamometers will also be conducted. A detailed analysis of suitable blowers and flow control methods is presented in Section 5.5 and 5.6 of this task report, and recommendations are made in Section 5.7.

5.2 REVIEW OF REQUIREMENTS

Paragraph 85.478-15 (b) of EPA's Proposed Rulemaking (NPRM) for New Motorcycles specifies the incorporation of a variable speed blower system for simulating engine cooling. The NPRM and the contract require that the blower have a mechanism, controlled by the dynamometer roll speed, which will regulate the blower flow to within 10 percent of the roll speed over an operating range of 10 kph (6.2 mph) to 100 kph (62 mph). At speeds less than 10 kph, the air flow shall be within ±1 kph of the simulated vehicle speed. The

NPRM and the contract also call out a minimum duct exit area of $0.5m^2$ and describe the relative positioning of the blower exit with respect to the motorcycle. The effects the mileage accumulation requirements would have on the design of the system are discussed in this report.

5.3 GENERAL SPECIFICATIONS

In light of the requirements for emissions tests as proposed in the NPRM and the contract, a basic set of minimum specifications can be established. These are listed in Table 5-1 and have been utilized to obtain from vendors design data on suitable blower systems. Systems which do not satisfy these specifications are also examined and consideration is given to the effects their operation might have on overall system performance and total cost.

Specifications for a blower system for the mileage accumulation tests would be identical, except the maximum linear velocity would increase to 30.56 m/sec (100.3 ft/sec) and the corresponding blower capacity would be 916 m^3/min (32,360 cfm).

5.4 DESCRIPTION OF EXISTING DYNAMOMETER/BLOWER SYSTEMS

A letter of inquiry was sent to all manufacturers of vehicle and motorcycle chassis dynamometers requesting literature on any motorcycle chassis dynamometers which they might produce. In that inquiry and subsequent follow-up conversations, information was obtained regarding engine cooling systems which are currently in use. The design of these systems vary. Most are constant-speed systems. Several are multi-speed systems, and a few can truly be classified as variable flow systems.

Table 5-1. GENERAL SPECIFICATIONS

Minimum linear velocity at blower outlet:	0.0 m/sec (0.0 ft/sec)
Maximum linear velocity at blower outlet:	27.78 m/sec (9.1 ft/sec)
Minimum blower outlet area:	0.5 m^2 (5.38 ft^2)
Maximum rate of change of linear velocity:	1.47 m/sec^2 (4.82 ft/sec^2)
Blower capacity (assuming blower outlet area of 0.5 m^2):	835 m^3/min (29,500 cfm)
Static pressure at duct exit:	0.00 Pa. (0.00 in. H_2O)
Noise level at maximum flow:	85 dBA
Response time of control system:	1 sec.
Degree of blower control (vehicle speeds >10 kph):	±10% of vehicle speed
Degree of blower control (vehicle speeds <10 kph):	±0.3 m/sec (0.9 ft/sec)

Burke E. Porter Machinery Company of Grand Rapids, has recently completed the installation of three motorcycle dynamometers in Kawasaki's new assembly plant in Lincoln, Nebraska. These units, designed for short period production line testing, are equipped with a 10 horsepower, 170 m^3/min (6,000 cfm), centrifugal blower. This single-speed system directs two jets of air towards the motorcycle and is designed to provide sufficient cooling for full power and high-speed testing of motorcycle engines up to 900 cc.

Hartzell Special Products, St. Paul, Minnesota, one of the larger manufacturers of motorcycle dynamometers, equips their units with single or dual blowers. Each blower has a capacity of 11.3 m^3/min (400 cfm) and an exit area less than 400 cm^2 (160 in^2). In the dual blower configuration, the cooling jet is directed tangentially across the side faces of the engine, while the jet from a single blower installation is directed at an angle across the engine. In both configurations, the exit of the blower is located just ahead of the front wheel.

The Super Dyno built by J & R Manufacturing, Bell Gardens, California, uses a small 3/4-horsepower blower, operating at a constant speed. In their design the blower is portable, permitting it to be placed in any position relative to the motorcycle. PABATCO, Athena, Oregon, also distributes a motorcycle dynamometer with a portable single-speed blower system.

Carl Schenck Maschinenfabrick, represented in the United States by Ostradyne Inc., produces two dynamometers for motorcycle testing. One is a sprocket unit while the other employs cradle rolls. These units, which are not factory equipped with a blower, require the user to install a blower which meets his respective requirements. Webco Santa Monica, California, a manufacturer of motorcycle parts, utilizes a two-speed blower with their Schenck dynamometer, and speed selection is controlled by a temperature switch, controlled by the engine head temperature. Kawasaki, also using a Schenck sprocket unit, uses a variable direction blower. As the engine speed changes, adjustments are made to the direction of the blower's guide vanes, changing the total fraction of air which passes the engine.

Akkerman Engineering and Manufacturing, Houston, manufactures a chassis dynamometer with a variable speed blower system. Their system employs a (14,000 cfm) fan, coupled to the hydraulic power absorber. As hydraulic pressure is being generated by the hydraulic pump, most of the pressurized fluid goes through a hydraulic motor to operate the blower. As the motorcycle speed increases, higher fluid pressures are produced, forcing more fluid through the fan's hydraulic motor, and increasing fan speed. The exit area of the blower is in excess of $0.5m^2$ and is positioned directly in front of the motorcycle. However, no attempt is made to equate the cooling air speed and the speed of the motorcycle.

5.5 BLOWER CHARACTERISTICS

There are several types of blowers which might be suitable for incorporation into the variable flow blower cooling system. Each of these types will be considered separately, and a number of the following blower design/ operational factors will be examined for each class of blowers. These factors include:

- Type of fan
- Housing Characteristics
- Impeller Characteristics
- Discharge Air Pattern
- Efficiency

- Air Flowrate vs. Pressure Drop
- Noise Level at Maximum Output
- Size/Configuration
- Duct Work Requirements
- Cost
- Delivery

In the following pages, the requirements for the emissions certification test, as they relate to engine cooling needs, will be discussed. The requirements for the mileage accumulation test will be similar; only the magnitude of the maximum airflow will be different.

5.5.1 Terminology

The following terms and symbols are used during the discussion of the blowers, and control systems.

Total Pressure (TP) - the air pressure resulting from passage of air through the blower

Static Pressure (SP) - the total pressure rise reduced by the velocity pressure in the fan

Brake Horsepower (BHP) - the horsepower imparted to the blower via the shaft of the fan wheel

Air Horsepower - fan power output determined from the product of air volume and total pressure

Mechanical Efficiency (ME) -	the ratio of air horsepower to brake horsepower
Static Efficiency -	the ratio between static pressure and the horsepower input

5.5.2 Centrifugal Blowers

There are four types of centrifugal blowers, each utilizing a different impeller design. The four impeller designs use:

- Airfoil blades
- Backward inclined/backward curved blades
- Radial and curved radial blades
- Forward curved blades

Each of these types has a different set of performance characteristics and noise emissions, and each has been designed for different applications. Design and performance data for all types of centrifugal blowers have been solicited from manufacturers. Data on centrifugal units which meet the specifications outlined in Table 5-1 and the above criteria have been tabulated in Table 5-2.

The characteristics of this generic class of blowers are described below in Section 5.5.2.1. Specific characteristics resulting from the impeller design are described separately.

5.5.2.1 General Description

In this class of blowers, the air pressure results from the centrifugal force created by rotating the air

Table 5-2.

CHARACTERISTICS OF CENTRIFUGAL BLOWERS

MANUFACTURER	IMPELLER TYPE	MODEL	INLET DIA. (m)	OUTLET HT (m)	SIZE WIDTH (m)	OUTLET AREA (m^2)	AIR FLOW (m^3/min)	SP = .75 kPa (3" H_2O)		SP = 1.0 kPa (4" H_2O)		EXPECTED SP (kPa)	MASS (kg)	APPROX. SIZE			PRICE ($) F.O.B. MFG F.O.B. Olson*	NOTES
								RPM	BPH	RPM	BHP			HEIGHT (m)	WIDTH (m)	DEPTH (m)		
Aladdin Heating	Bckwrd. Incl.	50BR	0.737	0.714	0.619	0.442	856	788	46.7	831	52.7	0.70	1202	2.080	1.810	1.820	-	Belt Drive
		57BR	0.838	0.813	0.702	0.570	840	587	33.8	641	40.5	0.70	1515	2.356	2.089	2.064	-	Belt Drive
		64BR	0.940	0.908	0.787	0.715	846	485	29.4	527	35.2	0.70	1894	2.626	2.286	2.288	-	Belt Drive
	Radial	50AR	0.737	0.714	0.619	0.442	856	785	46.7	828	57.2	0.70	1202	2.080	1.810	1.820	-	Belt Drive
		57AR	0.838	0.813	0.702	0.570	840	585	36.4	628	42.1	0.70	1515	2.356	2.089	2.064	-	Belt Drive
		64AR	0.940	0.908	0.787	0.715	846	469	29.6	510	35.6	0.70	1894	2.626	2.286	2.288	-	Belt Drive
Chicago Blower	Airfoil	36-1/2 SISW	-	0.898	0.806	0.724	839	1159	27.8	1206	32.2	0.75	-	1.810	1.608	1.633	1622	Belt Drive
		40-1/4 SISW	-	0.997	0.895	0.893	859	929	24.1	979	28.7	0.80	-	2.007	1.637	1.784	2124	Belt Drive
		44-1/2 SISW	-	1.102	0.990	1.092	853	728	20.5	793	25.4	0.85	-	2.210	1.775	1.964	2469	Belt Drive
		30 DIDW	-	0.708	1.216	0.861	840	1245	23.0	1314	27.7	0.95	-	1.638	1.591	1.334	1756	Belt Drive
		33 DIDW	-	0.784	1.338	1.049	832	1000	19.8	1070	24.4	1.00	-	1.810	1.759	1.465	1770	Belt Drive
Barry Blower	Bckwrd. Incl.	365 SISW	-	0.983	0.733	0.720	868	1196	31.6	1248	36.2	0.70	450	1.745	1.073	1.537	-	Direct or Belt
		402 SISW	-	1.081	0.814	0.880	844	930	24.6	986	29.3	0.75	599	1.932	1.168	1.694	-	Direct or Belt
		445 SISW	-	1.198	0.895	1.072	839	747	20.9	806	25.8	0.80	748	2.129	1.273	1.832	3170*	Direct or Belt
		270 DIDW	-	0.727	0.973	0.707	854	1574	31.4	1649	36.1	0.85	319	1.307	1.340	1.135	-	Direct or Belt
		300 DIDW	-	0.808	1.078	0.871	844	1224	24.9	1304	29.8	0.90	444	1.448	1.403	1.265	3330*	Direct or Belt
New York Blower	Airfoil	449 SISW	1.130	1.260	0.851	1.072	839	825	18.6	884	23.4	0.80	974	-	-	-	2207	Direct or Belt
		339 DIDW	0.838	-	-	1.047	830	1100	19.6	1184	24.8	0.80	513	-	-	-	1740	Direct or Belt
	Bckwrd. Incl.	448	1.130	1.260	0.851	1.072	839	757	20.0	820	25.2	0.80	975	-	-	-	2220	Direct or Belt
	Curved Radial	784 DH	1.143	1.127	0.937	1.056	880	333	23.3	370	29.6	0.80	3513	3.400	2.505	3.245	7265	Belt Drive
	Radial	784 LS	1.143	1.127	0.937	1.056	880	336	26.4	375	33.5	0.80	3314	3.400	2.505	3.245	6605	Belt Drive

column between the blades and by the kinetic energy imparted to the air by virtue of its velocity leaving the impeller. The air velocity resulting from the rotative velocity is a result of the impeller design and the fan flow rate.

The majority of centrifugal blowers have single inlets (SISW), but some units utilize inlets on both sides of the blower. These latter units, denoted as DIDW units, are typically smaller than SISW units having similar characteristics. As an example, Chicago Blower's Model 40¼ SISW and Model 30 DIDW are similar units. The outlet areas of the two units are almost identical, but the DIDW is smaller in size, operates at a higher impeller speed, and requires slightly less power. Similar comparisons can be made between the DIDW and SISW units fabricated by Barry Blower and New York Blower.

The centrifugal blower is driven directly by a motor coupled to the impeller shaft or indirectly by a motor connected to the shaft by a belt. The selection of direct-coupled motors or belt-driven motors is not always available to the user since many blower mounting configurations do not permit the use of direct-drive motors. When belt-drive motors are used, some slippage between the belt and shaft occurs, reducing the efficiency of the system. As a result of the frequent high acceleration and deceleration torque requirements, this slippage would be unacceptable if a variable-speed motor were used to attain the variable flow control. Therefore, in order to obtain maximum efficiency and permit the usage of variable-speed motor control systems, only blower mounting arrangements which permit the use of direct-drive motors will be considered.

The fan casing, commonly called the scroll or spiral, collects the air delivered from the impeller and directs the flow into the exit duct. As a result of the curvature of the scroll and the centrifugal action of the impeller, the resultant velocity profile at the exit of the blower is nonuniform. In most standard applications, this nonuniformity is not a problem, but it may not be acceptable for this application. Directing vanes can be used to transition the spiral flow in the housing to streamline flow in the exit duct. These vanes can also be used to restructure the velocity profile, producing a more uniform profile.

Typically, the exit configuration of centrifugal fans is rectangular in shape and a scroll configuration can be selected that would result in the outlet duct being positioned just above and parallel to the floor. However, even though the exit duct is appropriately located with respect to the motorcycle, some additional ducting is required. This will include telescoping ducting in order to position the duct exit just in front of the front wheel of the motorcycle; flow straightening and conditioning sections; and a transition section from the blower exit duct dimensions to the required exit area of at least $0.5m^2$ (5.4 feet^2).

The pressure loss at maximum flow in this ducting can be approximated by the following equation:

$$H_d \text{ (kPa)} = 0.45 + 0.45\ C_1 + 0.03n + H_S \qquad (5\text{-}1)$$

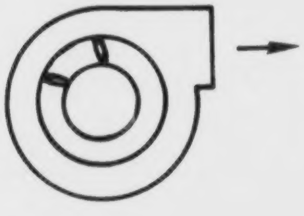
The first term accounts for the pressure drop which occurs at the exit of the duct when the velocity pressure of the cooling air is converted to static pressure. The second term describes the pressure drop associated with a change of duct area from the exit area of the blower duct to the final exit area of at least $0.5m^2$. The coefficient C_1, is a function of the ratio of area of the exit blower duct to the final exit area of $0.5m^2$. The magnitude of C_1 will also depend on whether the transition is gradual or abrupt. The next term accounts for the losses attributable to the straightening vanes, where n is the number of vanes. The final term H_S is the pressure loss across a noise baffle, if it should be used. Considering all of these terms, it is anticipated that the total pressure drop would be 0.50 - 0.88 kPa (2 - 3½ in. H_2O). Approximate values for the pressure drop for the various blowers considered are tabulated in Table 5-2.

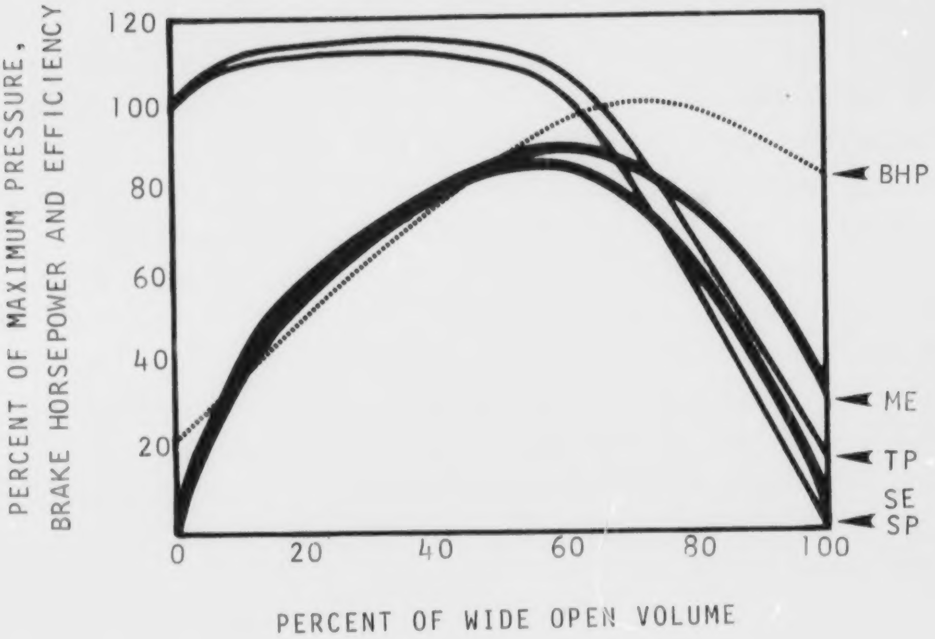
Should a centrifugal blower be used for this application, it would have to be firmly mounted to the floor of the test cell. This would be necessary because of the mass of the blower, which, excluding motor and controls, typically weighs more than 900 kg (2,000 lbs), and the effect vibrations might have on the mechanical balance of the impeller wheel and, thus, the performance of the blower system. The blower would be mounted so that the exit duct would be centered about the front wheel restraints of the dynamometer, thereby providing symmetric cooling to the motorcycle engine. A similar mounting scheme could be used on a system for testing three-wheeled motorcycles.

The fan wheel inertia is also an important parameter, particularly when a variable-speed motor control is used. The relationship between motor size and the wheel inertia is discussed in Section 6. In centrifugal blowers the magnitude of the fan wheel is substantial. The inertia of New York Blower's Size 449 and 448 units is just under 25 kgm^2 (600 lb ft^2) while equivalent DIDW units are approximately one-half that size.

As shown in Figures 5-1 to 5-4, the sound power levels from a blower operating at the anticipated conditions leave the fan wheel at a velocity slower than the impeller tip speed. Also as a result of the blade depth, efficient are substantial. Any reductions in these levels would reduce the requirements for sound insulation and provide a better working environment for the vehicle driver. Substantial noise reductions can be obtained by utilizing a fan silencer. These devices are commercially available and can be attached directly to the inlet or outlet of the blower. The reduction in sound power levels is shown in Table 5-3. This magnitude of reduction can be achieved in all types of blowers, not just centrifugal units.

IMPELLER CHARACTERISTIC





BLOWER CHARACTERISTICS

120
100
80
60
40
20
0
20
40
60
80
100
PERCENT OF MAXIMUM PRESSURE, BRAKE HORSEPOWER AND EFFICIENCY
PERCENT OF WIDE OPEN VOLUME
BHP
ME
TP
SE
SP

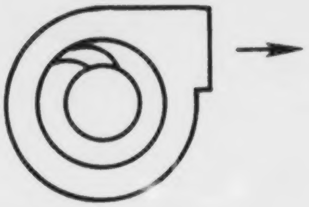
SPECIFIC SOUND POWER LEVEL

dB re 10^{-12} watt, $849 m^3/min$ @ 1 kPa Fan Total Pressure

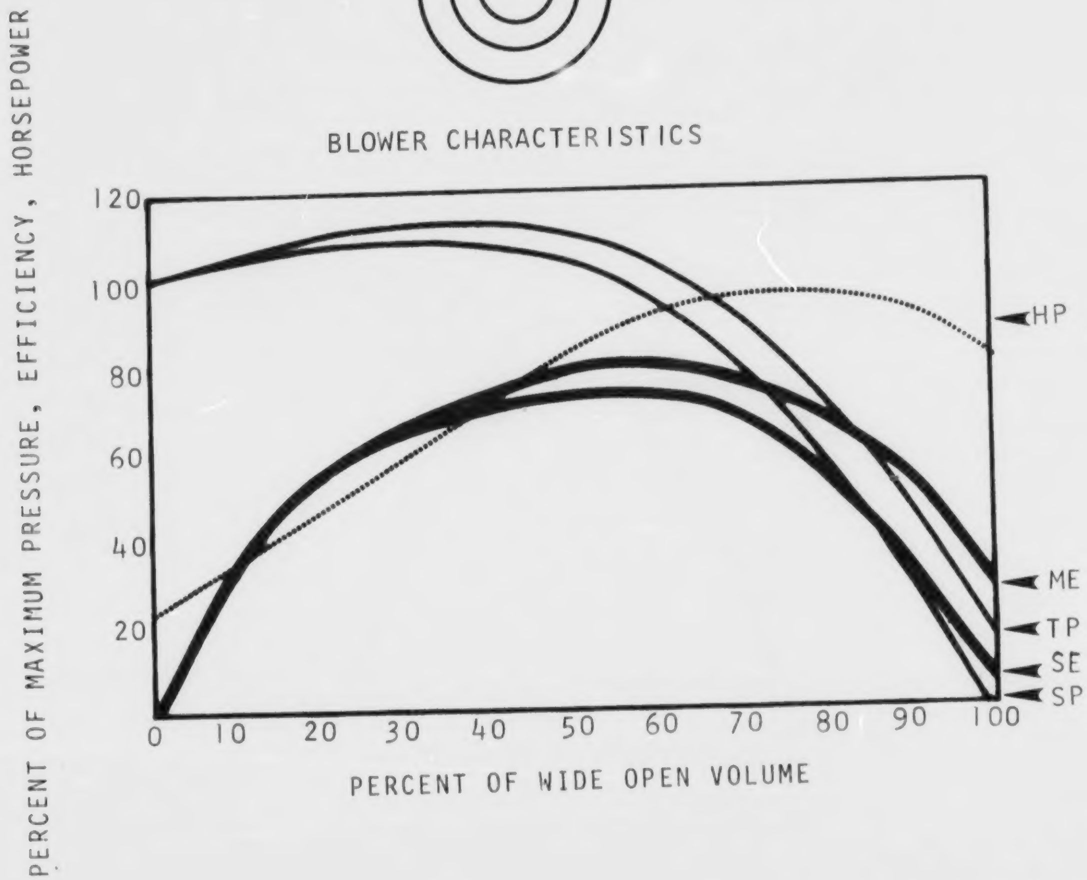
OCTAVE	1	2	3	4	5	6	7	8
CENTER FREQ	63	125	250	500	1000	2000	4000	8000
LEVEL	92	92	91	89	88	83	75	67

Figure 5-1. CENTRIFUGAL BLOWER AIRFOIL IMPELLER

IMPELLER CONFIGURATION



BLOWER CHARACTERISTICS



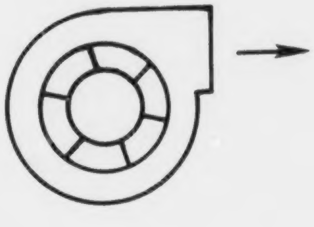
SPECIFIC SOUND POWER LEVEL

dB re 10^{-12} watt, $849 m^3/min$ @ 1 kPa Fan Total Pressure

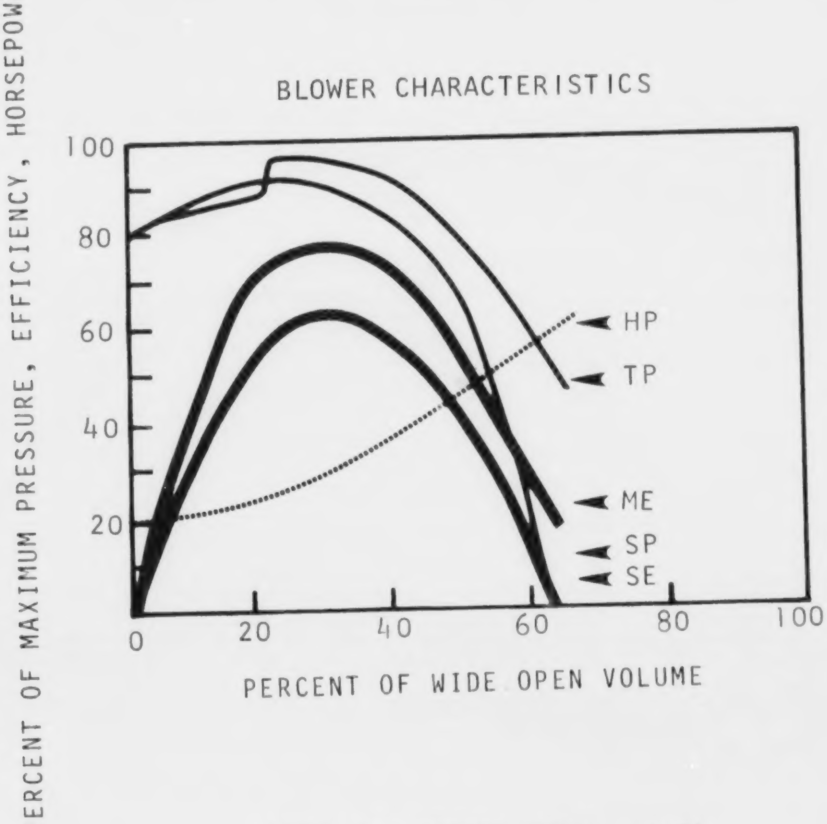
OCTAVE	1	2	3	4	5	6	7	8
CENTER FREQ	63	125	250	500	1000	2000	4000	8000
LEVEL	92	92	91	89	88	83	75	67

Figure 5-2. CENTRIFUGAL BLOWER
BACKWARD INCLINED/BACKWARD CURVED IMPELLER

IMPELLER CONFIGURATION



BLOWER CHARACTERISTICS



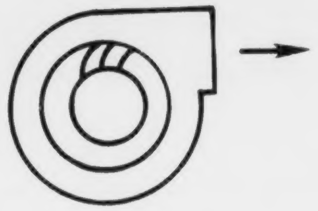
SPECIFIC SOUND POWER LEVEL

dB re 10^{-12} watt, $849 m^3/min$ @ 1 kPa Fan Total Pressure

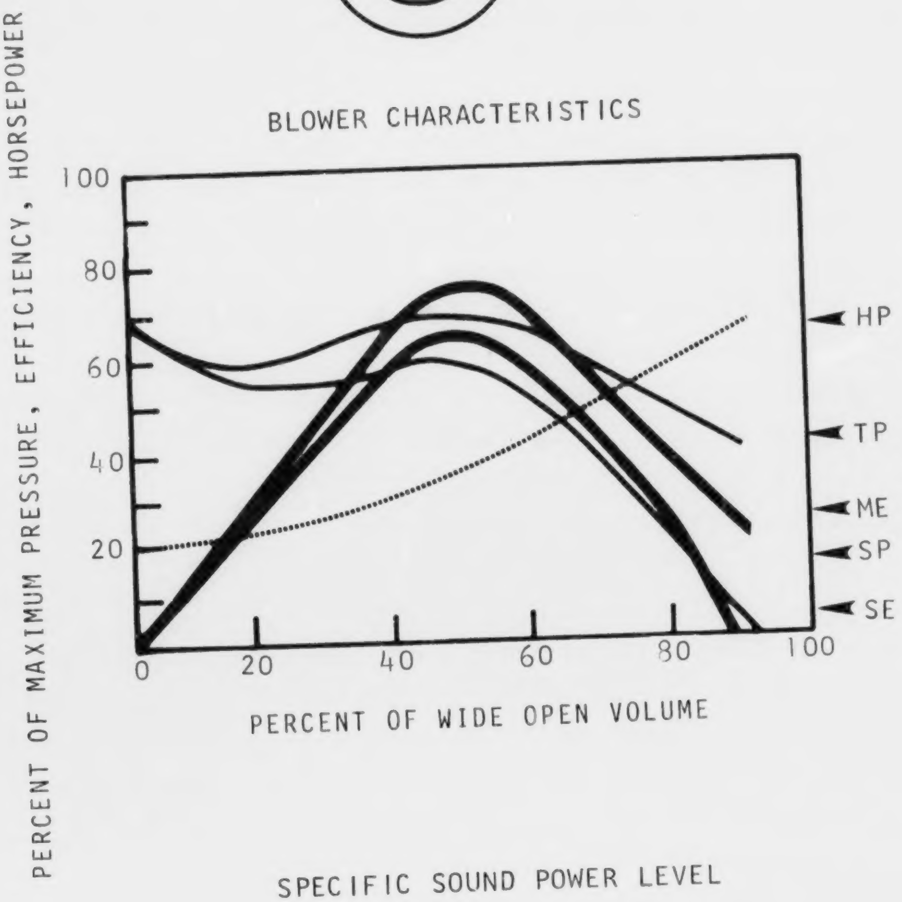
OCTAVE	1	2	3	4	5	6	7	8
CENTER FREQ	63	125	250	500	1000	2000	4000	8000
LEVEL	105	102	100	100	95	90	87	86

Figure 5-3. CENTRIFUGAL BLOWER RADIAL IMPELLER

IMPELLER CONFIGURATION



BLOWER CHARACTERISTICS



SPECIFIC SOUND POWER LEVEL

dB re 10^{-12} watt, 849m^3/min @ 1 kPa Fan Total Pressure

OCTAVE	1	2	3	4	5	6	7	8
CENTER FREQ	63	125	250	500	1000	2000	4000	8000
LEVEL	97	95	95	91	85	81	78	72

Figure 5-4. CENTRIFUGAL BLOWER
FORWARD CURVED IMPELLER

Table 5-3. TYPICAL SOUND POWER LEVEL
REDUCTIONS BY FAN SILENCERS

Octave Band		1	2	3	4	5	6	7	8
Center Frequency		63	125	250	500	1000	2000	4000	8000
	Velocity								
Dynamic	0	3	6	16	23	25	23	18	16
Insertion	100 m/sec	2	5	15	22	24	22	17	15
Loss (dB)	300 m/sec	2	4	14	21	23	21	16	14

There are several penalties associated with the installation of a fan silencer. The most obvious penalty is the cost of the unit. Also, as a result of installing the silencer on the blower outlet, a substantial pressure loss is incurred. The magnitude of that loss can almost double the total system pressure drop and increase the horsepower requirements by as much as 20 percent. However, since the silencer may also serve as a mixing baffle, its installation may improve the uniformity of the air exiting from the duct.

5.5.2.2 Airfoil Impellers

In this design, the impeller, which is shown in Figure 5-1, consists of ten to sixteen airfoil blades curved away from the direction of rotation. This causes the air to leave the fan wheel at a velocity slower than the impeller tip speed. Also as a result of the blade depth, efficient expansion of the air occurs within the blade passages. These features result in a blower design which is highly efficient and relatively quiet.

The performance characteristics are also shown in Figure 5-1. While the sound power levels are high, they are lowest of all the fans examined in this report. The performance curves show that the peak efficiency occurs when the fan is operating at 50 to 60 percent of free flow. The peak efficiency of 80 to 85 percent is higher than that obtained by other types of blowers. This also is an operating area where the pressure curve is rising, thus providing good operational stability near the point of peak power consumption. The static pressure trace has been normalized so that the 100 percent point is at the point of maximum efficiency.

As seen in Table 5-2, the airfoil designs, built by New York Blower and Chicago Blower, require less power to reach the desired flow condition than do the blowers equipped with backward inclined or radial blades. However, in order to take advantage of this design's inherent efficiencies, it is necessary to go to a larger scroll and fan wheel size. This results in a larger blower system, and substantially larger exit duct areas which, in turn, result in a larger pressure drop when transitioning down to the exit duct area of $0.5m^2$ ($5.4\ ft^2$). Relative costs for these blowers are also shown in Table 5-2.

5.5.2.3 Backward Curved/Contoured Blades

This impeller design, which is sketched in Figure 5-2, is very similar to the airfoil design. The only real difference is that the blades are of uniform thickness and backwardly curved or inclined. As seen in the performance curves, this results in a slight reduction in efficiency, 1 to 5 percent, and a negligible increase in sound output, as compared to the airfoil design.

As shown in Table 5-2, several blowers of this type are available which would only require a minimal exit duct transition piece. However, as seen in the specifications of the Aladdin units (Table 5-2), a severe horsepower penalty is incurred by going to the unit with the smaller exit area. The smaller unit is slightly less expensive, but the costs for the smaller motor required by the larger unit may offset this price differential, and result in a lower net price for the larger system.

5.5.2.4 Radial Blades

In this design, six to ten blades are radially mounted on the impeller. The blades may be as shown in Figure 5-3, or they may be slightly curved. The resultant design has high mechanical strength and is easily repaired. It has been extensively used in material handling applications where equipment ruggedness and design simplicity are essential. The size, weight, and cost of these blowers is substantially greater than airfoil or backwardly inclined blowers having comparable characteristics. The data on the New York Blowers, tabulated in Table 5-2, shows the relationships between the three styles of blowers.

As shown in the performance curves in Figure 5-3, this style of blower has substantially higher pressure characteristics than the other types of blowers, but the radial type has a lower efficiency throughout its entire range of operation. The sound power output levels for this style are also the highest of all the blowers being considered. In contrast to the previously described centrifugal blowers, the horsepower curve for radial blowers shows a continual rise to a maximum which occurs at free flow.

5.5.2.5 Forward Curved Blades

Blowers equipped with forward curved blades are used primarily in applications such as residential heating and ventilation units and packaged air conditioning equipment which requires low pressure and small to moderate air volumes. The performance curves and sound characteristics of this style are displayed in Figure 5-4.

Vendors did not supply any data on this type of blower, because it is incapable of meeting this application's air volume requirements. Therefore, further consideration of this style is unwarranted.

5.5.3 Axial Blowers

Axial blowers are characterized by the mechanism in which they produce velocity and pressure changes. In this class, air pressure is produced only as a result of forward velocity changes which occur when air passes through the fan wheel.

Three types of blowers can be classified as axial units. These are:

- Propellers
- Tubeaxial Fans
- Vaneaxial Fans

The three types of axial units have similar design and performance characteristics which are discussed below in Section 5.5.3.1. The characteristics specific to each style are considered separately.

Data on axial units, which meet the requirements outlined in Table 5-1, have been solicited from vendors, and have been tabulated in Table 5-4.

5.5.3.1 General Description

In contrast to centrifugal blowers, axial blowers have both a circular inlet and outlet, usually of comparable areas. In many applications, an entrance orifice is installed at the blower inlet. This curved inlet addition reduces the entrance pressure loss on installations where no duct work precedes the fan. Since the outlet of the axial blower is circular, ducting will be required to transition the air from this circular cross section to the rectangular configuration required at the exit. However, the pressure drop which results in this geometry transition is no greater than that incurred in the reduction of a centrifugal blower exit area from one rectangular size to another.

Table 5-4.

CHARACTERISTICS OF AXIAL AND MIXED AXIAL BLOWERS

MANUFACTURERS	TYPE	MODEL	INLET DIA. (m)	OUTLET DIA. (m)	OUTLET AREA (m^2)	RPM	SP = 0.75 kPa (3" H_2O)		SP = 1 kPa (4" H_2O)		EXPECTED SP (kPa)	MASS (kg)	APPROX. SIZE		COST ($)	NOTES
							AIR FLOW (m^3/min)	BHP	AIR FLOW (m^3/min)	BHP			MAX DIA. (m)	LENGTH (m)		
Aerovent	Vaneaxial	V311-Y42	0.967	0.846	0.562	1750	714	21.1	568	22.9	0.60	324	0.967	0.762	-	Direct Drive
		V361-Y34	1.023	0.916	0.659	1750	814	23.7	731	25.0	0.65	395	1.023	0.838	-	Direct Drive
		V361-Y38	1.023	0.916	0.659	1750	903	29.7	816	30.4	0.65	426	1.023	0.838	-	Direct Drive
		V361-Y42	1.023	0.916	0.659	1750	980	34.2	888	34.7	0.65	474	1.023	0.838	-	Direct Drive
		V381-Y30	1.094	0.973	0.744	1750	854	22.9	753	24.1	0.65	442	1.094	0.838	-	Direct Drive
Hartzell	Vaneaxial	VA36A	1.041	0.914	0.656	1750	823	24.7	-	-	-	-	1.041	1.371	657	Direct Drive
		50 36VR3	0.914	0.914	0.656	1750	863	25.5	807	28.5	0.65	-	1.029	1.524	987	Direct Drive
Pacific Air Industries	Mixed Axial	270	0.944	0.944	0.700	1750	792	42.0	-	-	0.65	-	1.048	1.303	2010	Belt Drive

The pressure loss in systems using this class of fan can also be estimated by an equation identical to equation (5-1).

Physically, axial units are smaller and weigh less than comparable centrifugal units. In many cases, the weight difference is greater than 50 percent. As a result, the axial blower may be able to be mounted in such a manner that the entire blower can be repositioned so as to maintain a constant spacing between the exit of the air duct and the front wheel of the motorcycle. This would be an alternative to mounting the blower firmly to the floor of the test cell and utilizing telescoping ducts to maintain the required separation.

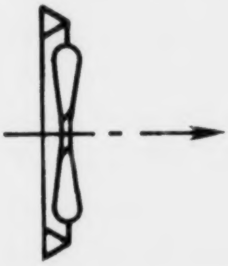
Axial blowers can also be operated by a motor directly attached to the fan or indirectly by a belt-driven system. Again, direct-drive systems are more efficient than the belt-driven system because slippages of the belt reduce the systems' efficiency. As concluded in the discussion of the centrifugal units, only axial units which permit the use of direct-drive motors should be considered because of the higher efficiency and adaptability to variable speed motors.

Fan silencers can also be used in conjunction with axial blowers, and sound reductions comparable to those shown in Table 5-3 can be achieved. However, most axial units would not be able to overcome the pressure drop introduced by the silencer. Flow straighteners could be used in conjunction with these units, but they would not appreciably reduce the noise emitted by these units.

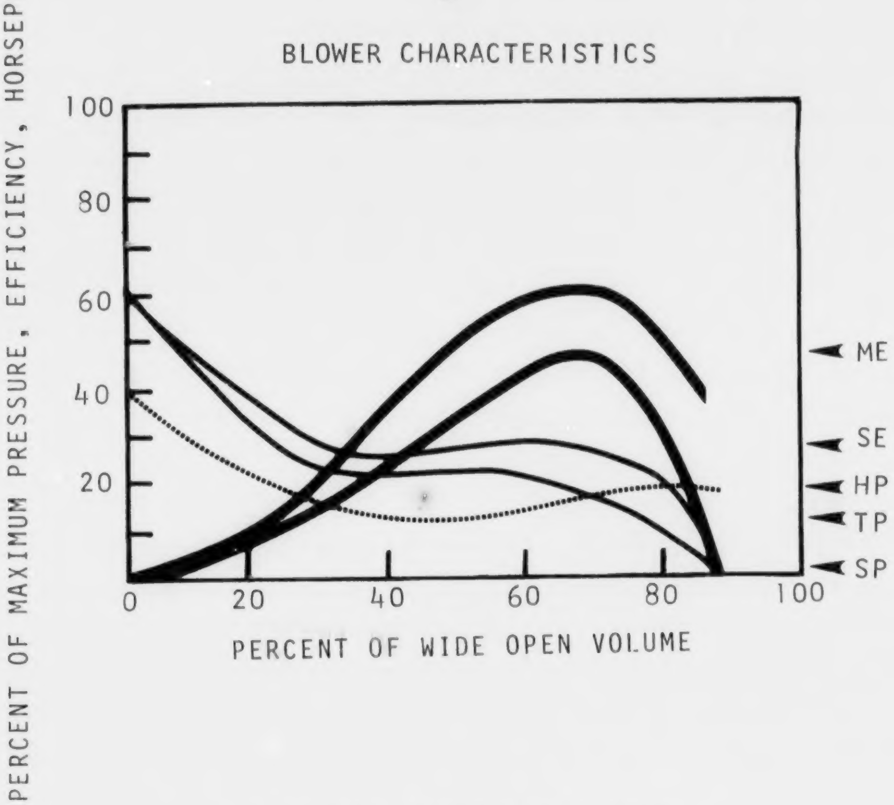
5.5.3.2 Propeller

The propeller is the simplest axial fan design and consists of two or more single thickness blades attached to a relatively small hub. This design, which is shown in Figure 5-5, is capable of creating high flow rates, but only

PROPELLER CONFIGURATION



BLOWER CHARACTERISTICS



SPECIFIC SOUND POWER LEVEL

dB re 10^{-12} watt, $849 m^3/min$ @ 1 kPa Fan Total Pressure

OCTAVE	1	2	3	4	5	6	7	8
CENTER FREQ	63	125	250	500	1000	2000	4000	8000
LEVEL	108	105	106	104	102	102	100	88

Figure 5-5. AXIAL BLOWER
PROPELLER

at low pressure levels. As seen in the performance curves, the efficiency of this style is very poor, and the sound power levels are very high. The discharge pattern of the air is circular in shape, and the airstream swirls as a result of the action of the blades and the lack of straightening vanes.

No propellers were identified which would be suitable for this application. Those which could meet the flow requirements would not be able to overcome the pressure drop which would be produced in the outlet transition duct and flow straightening section.

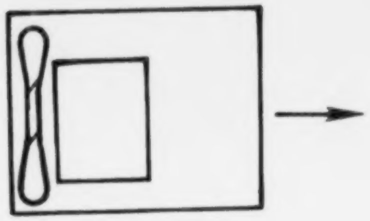
5.5.3.3 Tubeaxial Fans

As shown in Figure 5-6, a tubeaxial fan consists of a propeller or fan wheel mounted in a cylindrical tube. Clearance between the wheel tip and the inner wall of the tube is very small. The propeller fan typically consists of four to eight, single thickness or airfoil blades attached to the wheel hub. The blade pitch affects the blower characteristics.

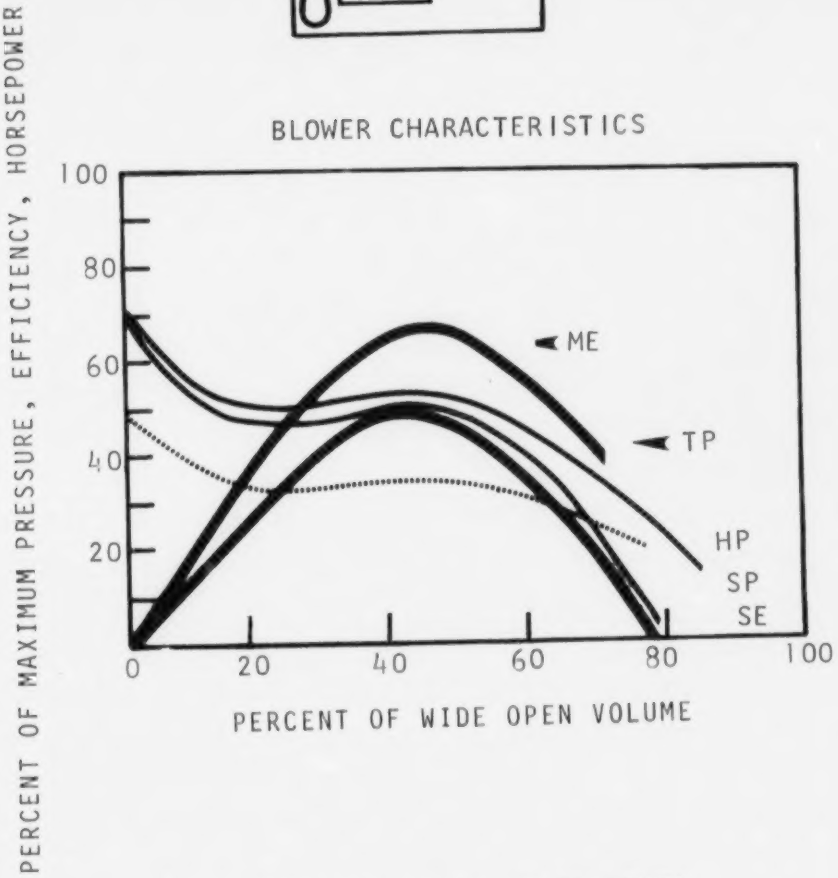
As a result of placing the propeller within the tube, improved efficiency at low and moderate pressure levels is obtained. This can be seen in the performance curves shown in Figure 5-6. These curves also show a dip in the pressure curve just to the left of the peak pressure point. This dip is caused by aerodynamic stall, and operation at this condition should be avoided. Resultant sound power levels are significantly lower than those for propellers, but higher than those for centrifugal blowers. The discharge air pattern from these units also exhibits a rotational or whirling motion which is a result of the rotation of the propeller and would require straightening vanes.

No information on suitable tubeaxial units was supplied by vendors.

TUBEAXIAL CONFIGURATION



BLOWER CHARACTERISTICS



SPECIFIC SOUND POWER LEVEL

dB re 10^{-12} watt, 849m^3/min @ 1 kPa FAN TOTAL PRESSURE

OCTAVE	1	2	3	4	5	6	7	8
CENTER FREQ	63	125	250	500	1000	2000	4000	8000
LEVEL	101	99	103	101	99	97	94	87

Figure 5-6. AXIAL BLOWER TUBEAXIAL

5.5.3.4 Vaneaxial Fans

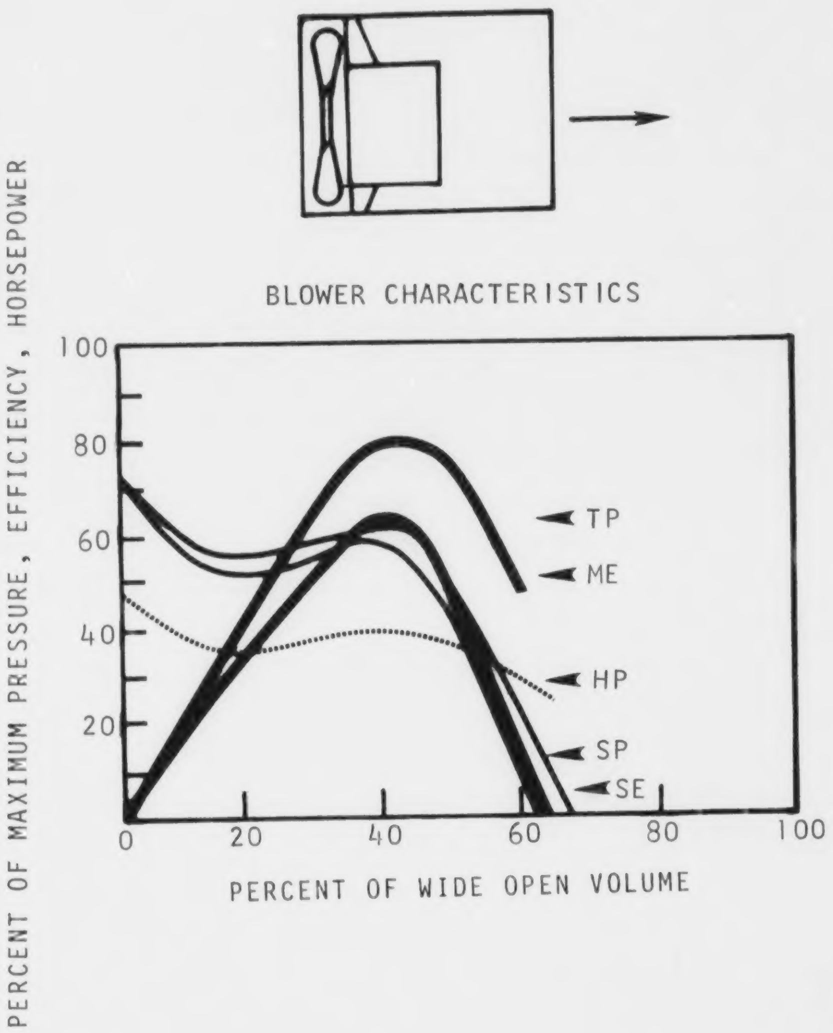
Vaneaxial fans are similar to tubeaxial units. The major difference is the inclusion of guide vanes just downstream from the fan wheel. The guide vanes straighten out the rotary motion imparted to the air by the rotation of the wheel. This improves the efficiency and pressure characteristics of the fan, as shown in Figure 5-7.

In order to achieve further improvement in these characteristics, airfoil blades are used and attached to a wheel hub which has a diameter greater than 50 percent of the wheel tip diameter. This further reduces the sound power levels, as shown in Figure 5-7.

The pitch of the blade is also an important characteristic as can be seen from the data tabulated in Table 5-5. The effects of pitch angle can be seen from the data on Aerovent's Series V 361 fans. The data indicates that by increasing the pitch from 86 cm (34 inches) to 107 cm (42 inches), the volumetric air flow at 0.50 kPa (2 inches H_2O) static pressure can be increased by almost 25 percent with a comparable increase in horsepower, but with no increase in the rotational velocity of the unit.

The rotational inertia of a vaneaxial fan wheel is less than 1/10th of that a centrifugal fan wheel having comparable performance. Therefore, although these units operate at higher wheel speeds per volume air flow delivered than centrifugal units, they will have higher acceleration rates and will require less power to accelerate the wheel.

In contrast to propellers or tubeaxial units, vaneaxial units are a viable means for satisfying the variable flow cooling problems of this application.



VANEAXIAL CONFIGURATION

BLOWER CHARACTERISTICS

SPECIFIC SOUND POWER LEVEL

dB re 10^{-12} watt, 849m^3/min @ 1 kPa FAN TOTAL PRESSURE

OCTAVE	1	2	3	4	5	6	7	8
CENTER FREQ	63	125	250	500	1000	2000	4000	8000
LEVEL	99	96	98	99	97	94	92	82

Figure 5-7. AXIAL BLOWER
VANEAXIAL

5.5.4 Mixed Axial Centrifugal Blowers

The mixed axial centrifugal blower combines the space saving features of the axial fan, i.e., 40 percent less installation space than centrifugal units, with the centrifugal blowers favorable performance characteristics, such as lower operating speeds and lower noise levels than associated with axial units. This is achieved by a unique air flow design which is shown in Figure 5-8. The air is drawn from the inlet into a centrifugal, backwardly inclined impeller and is discharged radially from the fan wheel. The air then must change direction by 90^0 where it enters the guide vane section.

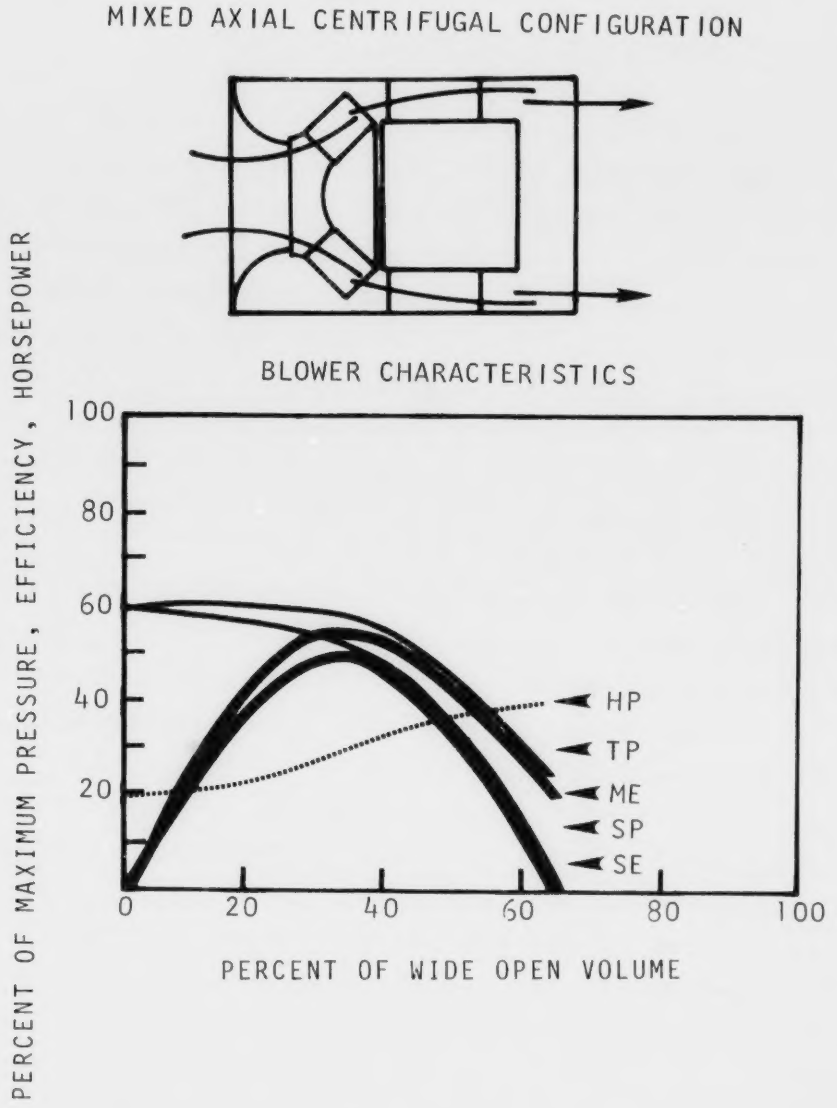
As a result, as seen in Figure 5-8, this fan's efficiency curve is comparable to a vaneaxial's, but slightly lower than the curve of a backwardly inclined centrifugal fan. Noise emissions are moderate, somewhat greater than a quiet centrifugal but lower than a vaneaxial unit.

The principal difference can be found in the pressure curves. The mixed axial's pressure curve shows a continually rising pressure curve characteristic, which permits the fan to operate throughout the pressure curve range without overload. Centrifugal or axial fans with peaking or reversing pressure curve characteristics cannot be operated to the left of the pressure peak because of air pulsations, excessive noise, and fan vibration.

Data on a suitable mixed axial centrifugal fan has been tabulated in Table 5-4.

5.6 VARIABLE FLOW CONTROL METHOD CHARACTERISTICS

The examination of a flow control system has been separated from the study of suitable blowers and fans because the control methodology is not dependent on the blower



SPECIFIC SOUND POWER LEVEL

dB re 10^{-12} watt, 849m^3/min @ 1 kPa FAN TOTAL PRESSURE

OCTAVE	1	2	3	4	5	6	7	8
CENTER FREQ	63	125	250	500	1000	2000	4000	8000
LEVEL	103	100	100	95	94	89	85	82

Figure 5-8. MIXED AXIAL CENTRIFUGAL BLOWER

design. Only when the control system is interfaced with the blower must the design parameters of the blower be explicity defined. In addition, no commercially available variable flow blower system has been identified. Rather, these systems are designed by the user to meet the specific requirements of the application.

As stated previously, a control system for the blower is required to equate the air speed at the exit of the blower ducting with the dynamometer roll speed. The control system should be capable of varying the blower flow rate over at least a 100-to-1 range. This large a range is required in order to meet the maximum flow rate requirement of 100 km/hr and the accuracy requirements of ±1 km/hr at dynamometer speeds less than 10 km/hr as specified in the control. An estimated response time of less than 1/4 second would be required to maintain the equivalence of air speed and vehicle speed within the accuracy limits specified in the contract. The exact degree of control and range of variable flow can best be specified after quantitative data have been obtained on the relationship of engine temperature and motorcycle exhaust emissions. Therefore, systems which meet either less stringent or more demanding specifications are examined in this report.

The discussion of the variable flow control methods has been divided up into two parts. In the first, the suitability of both open-ended and closed-looped systems will be examined. The other part of the discussion will closely examine the specific methods of achieving a variable flow control system for a blower.

5.6.1 Control Methods

Either an open-ended or closed-loop system can be utilized to control the variable flow blower system.

The open-ended system is shown schematically in Figure 5-9. In this system a change in motorcycle speed is sensed by the dynamometer tachometer. This signal is used to directly alter a precalibrated mechanism which modifies the blower flow. Since the linkage between the speed signal and the control mechanism is "hard," the response time is minimal and depends only on the electronic response time of the amplifiers and any control mechanism actuators.

The inherent simplicity of the open loop system is attractive. However, a mechanical or electrical interface is required unless the control mechanism is inherently linear. For most applications either mechanical or electrical linearization devices can be designed, but extensive design and testing is usually required to "time" the linearizer and establish system reliability because of the uniqueness of each system. The performance of the open-ended system will depend on the accuracy of any linearizer and on the repeatability and stability of the controller's cause and effect relationship.

In a closed loop system a feedback sensor is added, as shown in Figure 5-10. The closed-loop control system does not require linearization or calibration of the blower control mechanism, but linearization of the signal from the feedback sensor may be required. The selection of components for the feedback system is critical because their performance characteristics define the stability and response time of the control system. Specifically, if the feedback sensor is a tachometer mounted on a fan wheel shaft, changes in shaft speed will sensed almost instantaneously and there will be a negligible effect on the system response time. If a velocity sensor is used, the time required for the air column to reach the sensor, in addition to the sensor response time, may define the system response. Therefore, consideration should be given to the characteristics of available velocity sensors. The techniques which are available include:

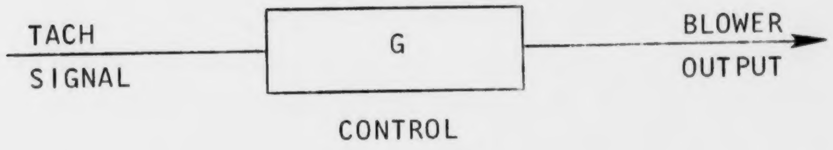

TACH
SIGNAL
G
CONTROL
BLOWER
OUTPUT

Figure 5-9. OPEN-ENDED CONTROL SYSTEM

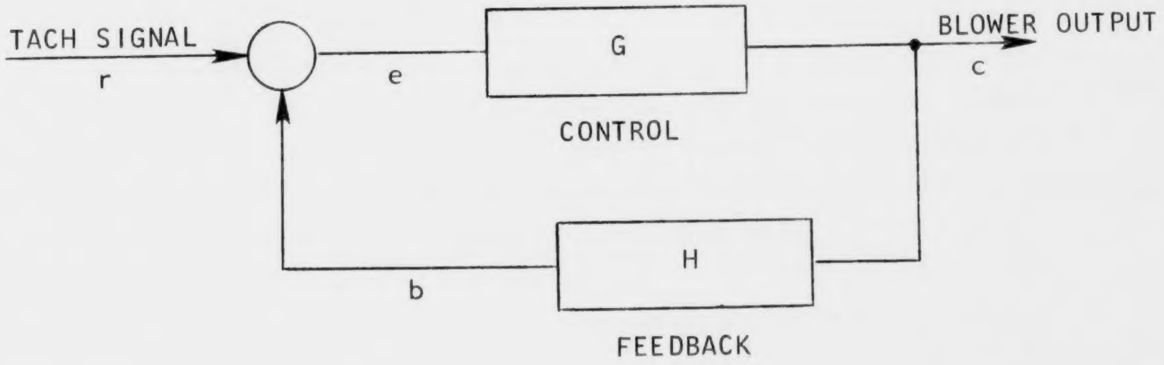

TACH SIGNAL
r
e
G
CONTROL
BLOWER OUTPUT
c
H
b
FEEDBACK

Figure 5-10. CLOSED-LOOP CONTROL SYSTEM

- Pitot tubes
- Turbine meters
- Heated thermopiles
- Hot wire anemometry

The pitot tube approach is extensively used in air pollution monitoring and investigations of heating/ventilating systems. It is accurate, usually within ±2 percent, and the velocity determination in this system is a function of the square root of pressure difference. Therefore, a linearization circuit would have to be developed.

Turbine meters are not suitable for this application because they cannot operate over the 100-to-1 dynamic range imposed by the contract specifications. Typically they function over a 20-to-1 range.

Heated thermopiles and hot wire anemometers utilize similar technologies to sense velocity. Since the parameter which is measured is the change of heat flux, the units tend to have a fast response time, in the range of 100-to-1,000 milliseconds, and accuracy within ±2 percent over a dynamic velocity range of 100-to-1. Inherently these sensors are nonlinear, but several manufacturers have developed electronic linearization circuits.

With any velocity sensor the positioning of the sensor and the design of the control loop are critical for minimization of instabilities and maximization of the control accuracy. The use of velocity feedback sensors is examined further in Section 6 of this report.

5.6.2 Control Mechanisms

Five blower control mechanisms which may be applicable are:

- Dampers
- Variable Speed DC Motor
- Constant Speed Motor with Eddy Current Clutch
- Variable Bypass Flow
- Compound Control System

Each of these mechanisms will be examined, and their advantages and disadvantages detailed. A summary of the findings has been incorporated into Table 5-5.

5.6.2.1 Shut-Off Dampers

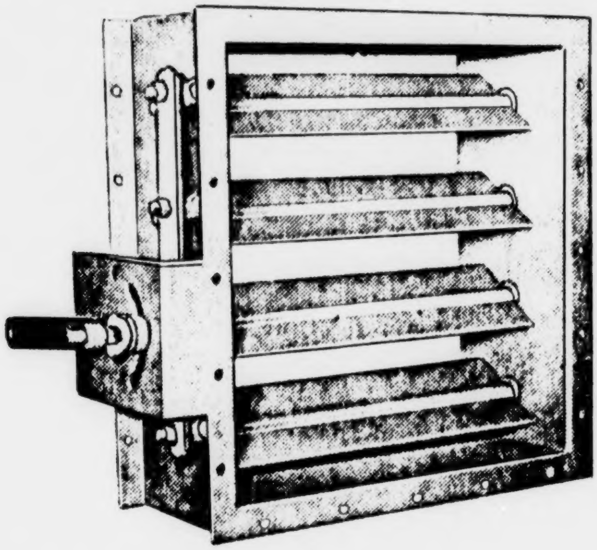
Dampers can be placed either on the inlet to the blower or at the exit. While the design of inlet and outlet dampers differ, the effect is the same; they restrict the amount of air a blower can deliver.

Outlet dampers are only used in conjunction with centrifugal blowers and are available in two different designs, parallel blade and opposed blade. The parallel blade design, shown in Figure 5-11, consists of four to six unidirectionally-oriented louvered vanes which effectively restrict the flow from the blower. As seen in Figure 5-11, the magnitude of the throttling action is a nonlinear function of the damper setting. As a result of the nonlinearity, this style is best suited for applications requiring accurate air volume control over a narrow range of flow; for example, from wide open to 75 percent of wide open capacity.

Opposed blade dampers, shown in Figure 5-12, utilize vanes which are oppositely louvered. This results in a slightly costlier design, but yields an almost linear control mechanism, and is, therefore, suited for control over a broad range of air volumes. However, operation at low flow rates is restricted because of the inherent instability of centrifugal blowers operating at low fractions of wide open capacity and the constant leakage through a

Table 5-5. SUMMARY OF CONTROL SYSTEMS

CONTROL SYSTEM (suppliers)	CONTROL METHOD CLOSED LOOP	OPEN LOOP	FEEDBACK SENSOR	REGULATION %	AIR FLOW RANGE MIN. (km/hr	MAX. (km/hr)	APPROX COST.	REMARKS
OUTLET DAMPER (Blower Mfgs)	X		Velocity	2%	15	100	$1500	
INLET DAMPERS (Blower Mfgs)		X	-	2%	15	100	$1500	
BYPASS - SINGLE DUCT (Olson Labs)		X	-	2%	0	100	$4000	System must be designed and tested.
BYPASS - DUAL DUCT (Olson Labs)		X	-	2%	0	100	$4000	System must be designed and tested.
EDDY CURRENT DRIVE (Eaton Dynamic) (Louis Allis)	X		Tach	1%	3	100	$6000	Dynamic braking may be required.
DC DRIVE (Eaton Dynamic) (Louis Allis) (Randtronics) (Sabina Electric) (Reliance Electric)	X		Tach	0.5%	1	100	$6800	Dynamic braking may be required.
REGENERATIVE DC-DRIVE (Randtronics) (Sabina Electric)	X		Tach	0.25%	0	100	$8000	



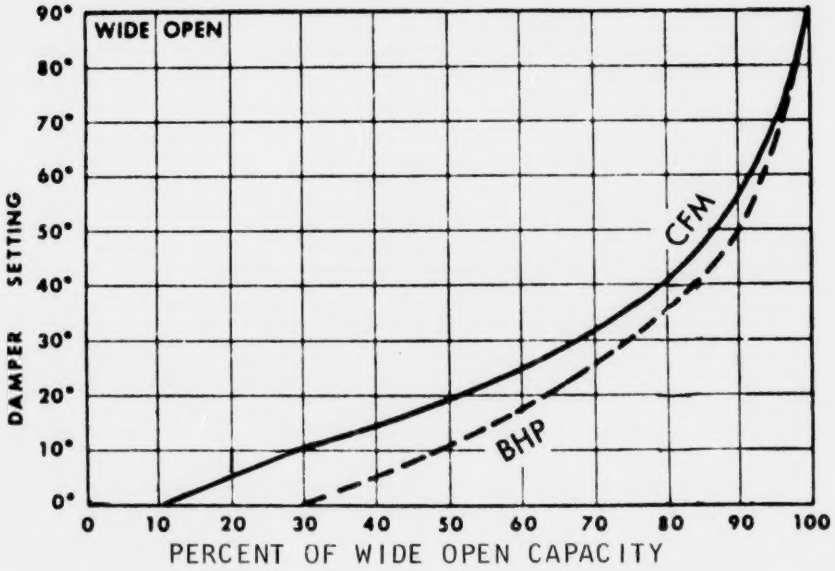


Figure 5-11. PARALLEL BLADE OUTLET DAMPER

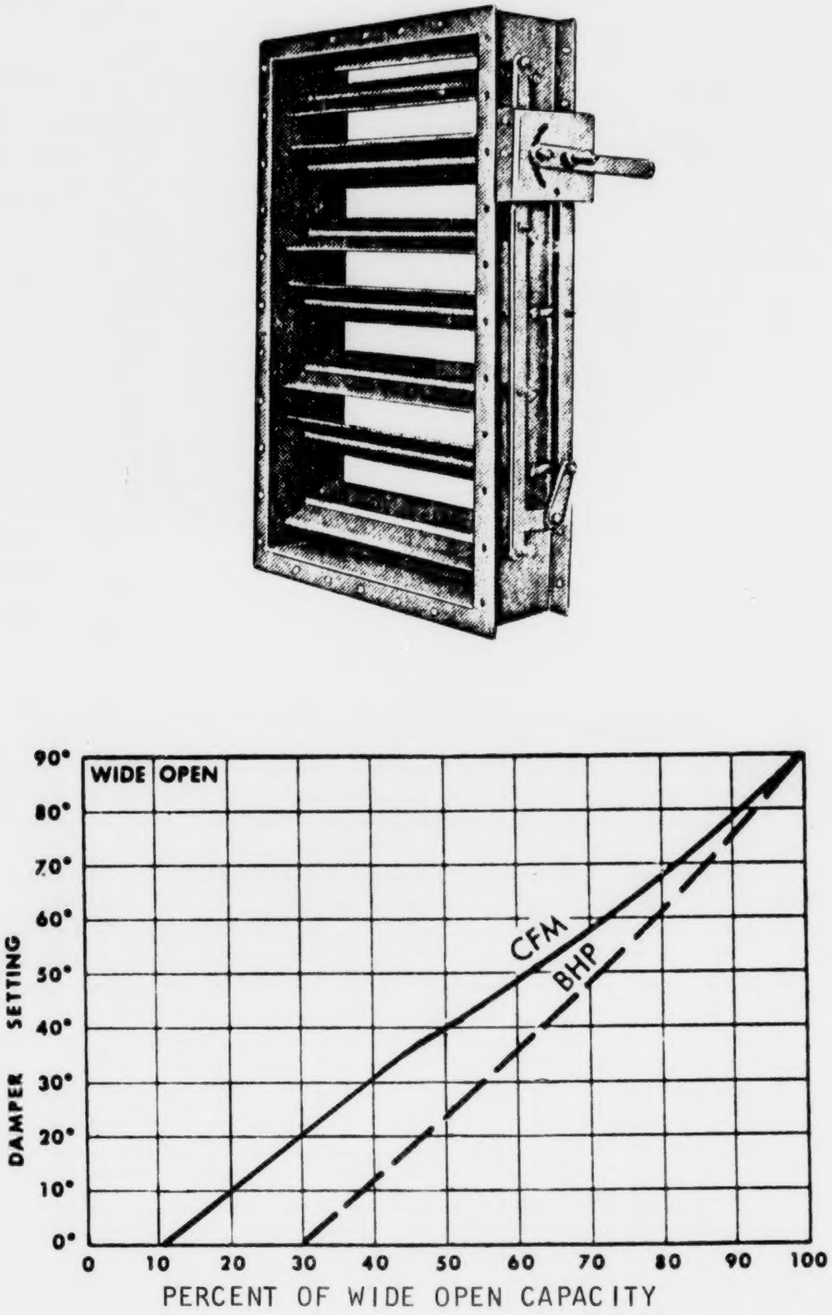


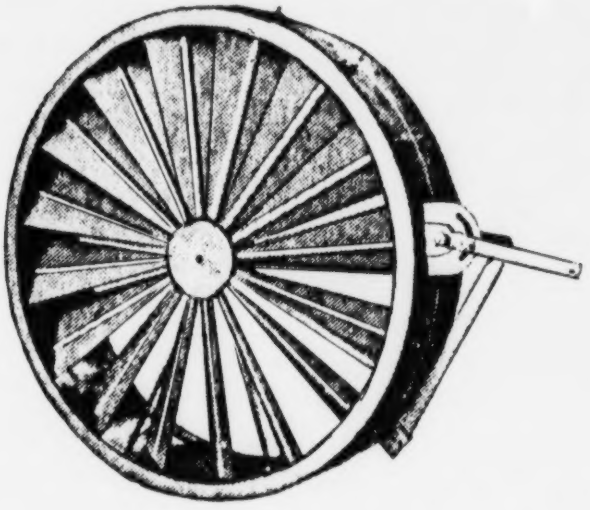
Figure 5-12. OPPOSED BLADE OUTLET DAMPER

supposedly closed damper. This leakage can be as high as 20 percent of wide open capacity.

An inlet vane control mechanism, as shown in Figure 5-13, is a nonlinear control device which is compatible with both centrifugal and vaneaxial blowers. This style of control provides more economical performance at lower air volumes than the outlet damper. The power savings for a centrifugal fan is graphically shown in Figure 5-14. The reduction in horsepower is achieved by using the inlet vanes to pre-spin the entering air in the same direction as the wheel rotation. Reduced operating costs at partial loads plus the use of the lowest cost, constant speed drive are additional advantages. However, even inlet dampers display significant leaks, approximately 15 percent of wide open capacity, under "shut-off" conditions.

As shown in Figures 5-11 to 5-13, the flow through the dampers is nonlinear with the degree of damper opening. However, a suitable linkage system can be designed so as to linearize this actuation. In addition, the damper vanes should be mounted in ball bearings rather than sleeve-type bearings to reduce friction and increase the positioning accuracy of the vanes. The positioning of the vanes is critical, and serious consideration should be given to the use of a closed-loop control if repeatable positioning of the dampers cannot be assured.

The leakage through either inlet or outlet dampers can be almost eliminated by using special felt-tipped vane-edge wiper seals. However when closed, the blower would be forced to operate in a "near shut-off" condition. When operated in this region, the blower, regardless of whether it is a centrifugal or a vaneaxial unit, may stall, vibrate and operate unstably. Therefore, as long as cooling air flows of less than 15 km/hr, are required, a shut-off damper control should not be considered. However, because of their low cost, $500 for damper, $1,000 to 2,000 for control



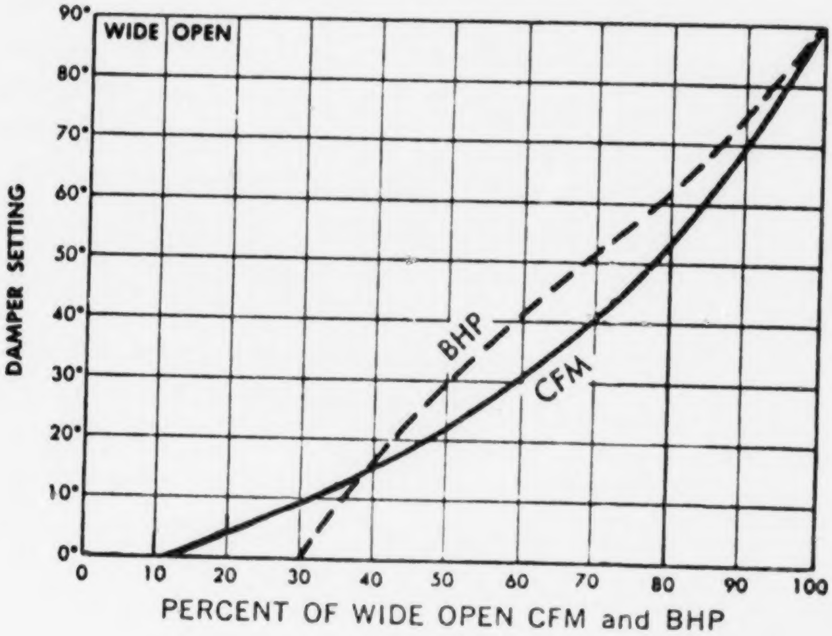


Figure 5-13. INLET DAMPER

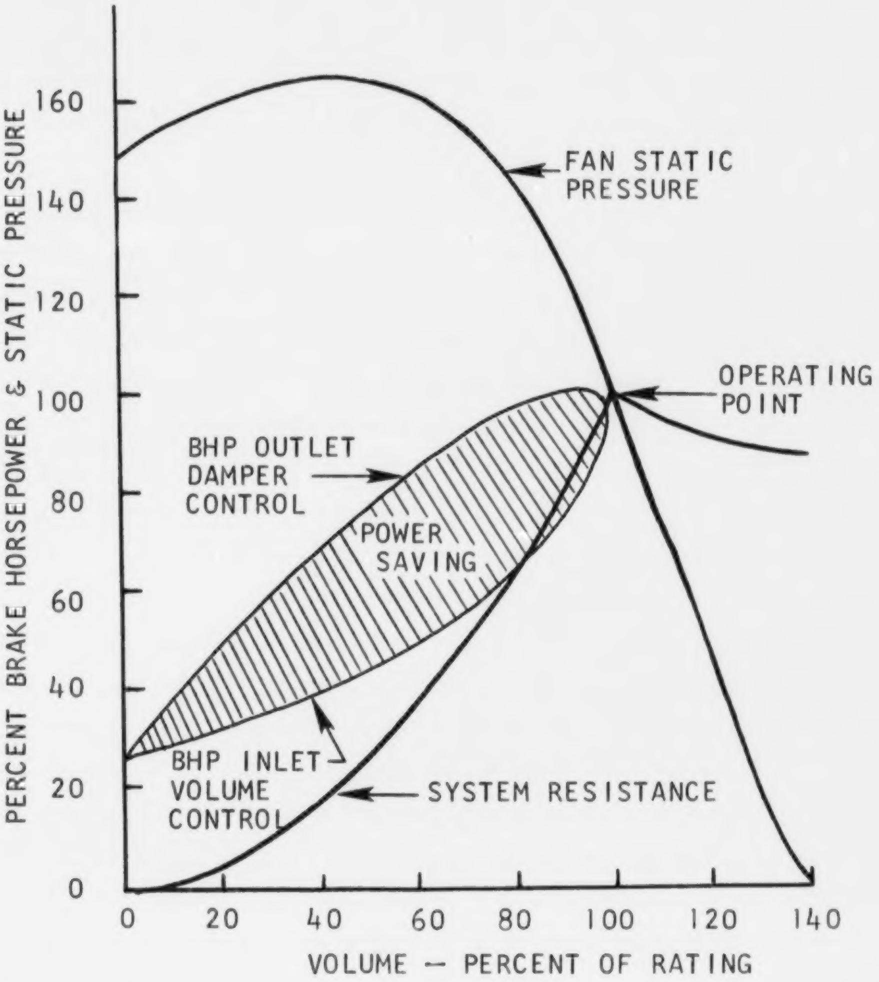


Figure 5-14. COMPARISON CURVES OF INLET AND OUTLET DAMPERS

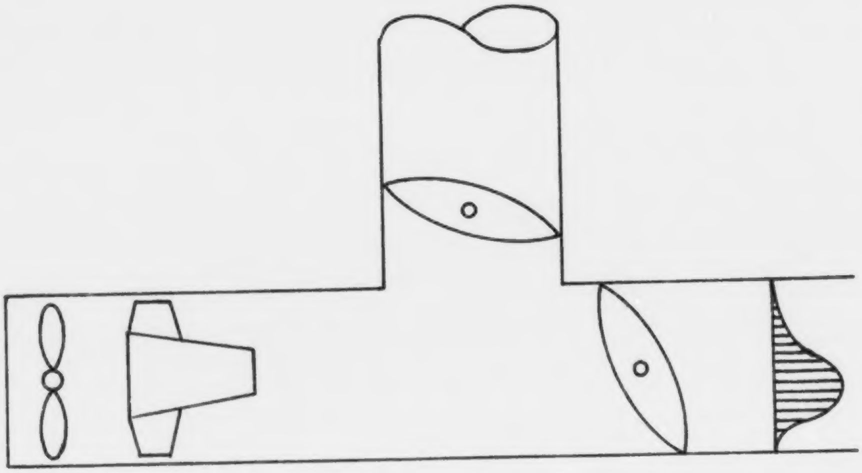
linkages and actuators, their use should be considered it flows less than 15 km/hr are not required.

5.6.2.2 Bypass Damper Flow Control

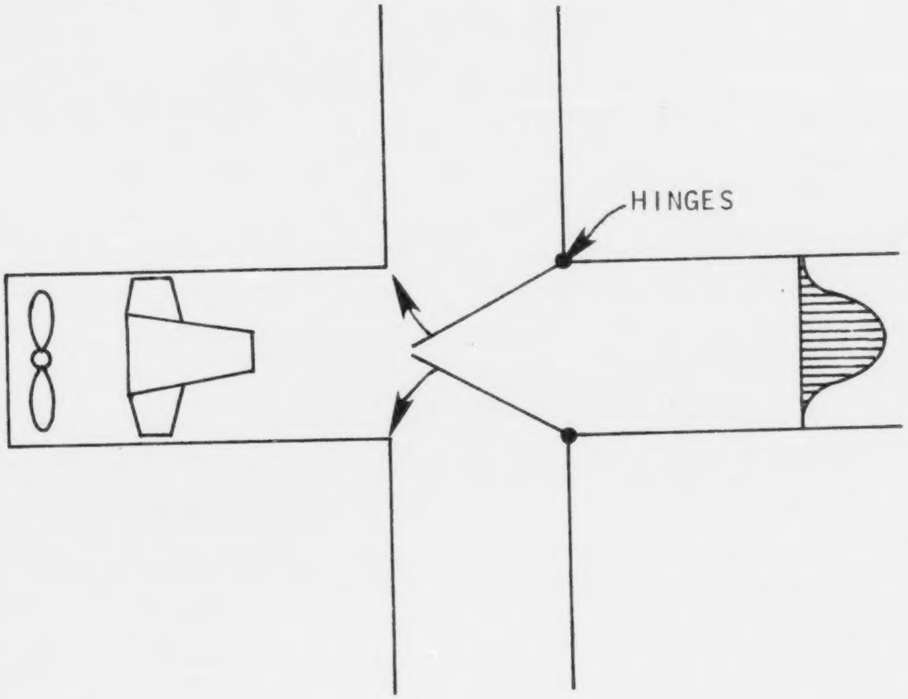
Two concepts of the bypass control method are shown schematically in Figure 5-15. In these configurations the fan is operated at a constant speed and constant static pressure, resulting in a constant flow. As a result, the fan can be operated at a condition of maximum efficiency. The velocity of the mainstream airflow is controlled by the balance between the mainstream outlet vane and the bypass outlet vane.

The bypass ducting splitter must be carefully designed because flow disturbances and distortions of the mainstream velocity profile can occur. This is particularly true if a single bypass duct is used as shown in Figure 5-15a. In this configuration the bypass air is required to make a 90° turn, and, as a result, the velocity profile of the mainstream air supply will be skewed as shown in Figure 5-15. Some smoothing of the flow profile can be achieved by using flow straighteners or a noise baffle, but the inclusion of these devices increases the pressure drop and the size of the system. Also, if a closed-loop control system with a velocity sensor is used, the system response time would be increased because of the added length of the ducting.

One configuration which minimizes the flow disturbances is shown in Figure 5-15. In this design, two bypass ducts are employed, and the flow is symmetrically divided about the center line of the ducting. Some distortions in the flow will still occur, and the resulting profile will be an exaggerated parabola. This effect can be reduced by utilizing an opposed blade damper whose multiple vanes will function as a flow mixer. Use of this style of damper would also result in a nearly linear control mechanism. Since the



a. SINGLE-DUCT BYPASS



b. DUAL-DUCT BYPASS

Figure 5-15. BYPASS VANE CONTROL SYSTEM

blower would always be operating at a constant condition, the dampers can be equipped with special vane-edge wiper seals in order to achieve the dynamic airflow range required by the specification.

Depending on the precision and linearity of dampers and related linkage system, this approach can be easily used in either a closed-loop or open-loop system. The cost of the system is estimated to be \$4,000 (\$1,000 for dampers and ducting, \$1,500 for controls and linkages, and \$1,500 for feedback sensor or precision controller).

There are several disadvantages to this design approach. First, the blower is always operating at its maximum capacity, which is three times the average required. According to the fan laws, horsepower varies as the cube of air volume, as shown in equation (5-2):

$$\text{new BHP} = \text{old BHP} \times \left(\frac{\text{new CFM}}{\text{old CFM}}\right)^3 \qquad (5\text{-}2)$$

Therefore, since the maximum LA-4 test speed is three times the average test speed a fan operating in a bypass configuration will consume approximately 27 times more power than a damper controlled or variable-speed controller system during a LA-4 cycle. In addition, in this design, two control mechanisms would have to be operated simultaneously, although they can be interconnected. Average sound emissions would also be high with this system since the blower is always operating at a peak condition. Finally, no suitable commercially available control vanes or valves have been located for this application, and the special design of suitable mechanisms could result in added design efforts and project costs.

5.6.2.3 Variable-Speed Drive Control

Variable-speed drive control is the most expensive means of achieving a variable-flow blower system. The costs for this system excluding a feedback sensor would be $5,000 to $8,500 depending on the size of the motor, the electronic braking system, and the system's stability and response time. However, this approach is proven and would be capable of satisfying the requirements outlined in Table 5-1.

According to the fan laws, which approximately define the characteristics of a fan operating under differing conditions,

$$\text{new CFM} = \text{old CFM} \times \left(\frac{\text{new RPM}}{\text{old RPM}}\right) \tag{5-3}$$

and

$$\text{new BHP} = \text{old BHP} \times \left(\frac{\text{new RPM}}{\text{old RPM}}\right)^3 \tag{5-4}$$

Since flow varies proportionately with rotational speed of the motor or fan wheel, a variable-speed motor control system is approximately a linear system which can be utilized in either open-ended or closed-loop control systems. Furthermore, as the rotational speed is being reduced, the required horsepower decreases cubically resulting in a substantial reduction in consumed power and component degradation. By controlling the RPM of the blower rather than its static pressure, operation of a blower under conditions of instability or stall is also avoided.

In a motor speed control system the motor selected must be able to follow the acceleration/deceleration rates outlined in the driving cycle in addition to the requirement of rotating the fan wheel at the maximum rotational speed.

The power required to achieve an angular acceleration rate is defined by the expression:

$$P = I A_r V_r \qquad (5\text{-}5)$$

where

I = rotational inertia of the fan wheel
A_r = rotational acceleration of the fan wheel
V_r = rotational velocity of the fan wheel

Since the rotational inertia of a vaneaxial blower is almost an order of magnitude less than that of a centrifugal blower's impeller, less power will be required to accelerate the former even though its angular acceleration rate is higher. It has been estimated that for a suitable vaneaxial blower, an additional 4 hp would be required to meet the acceleration requirements imposed by the driving cycle. An additional 10 hp would be required for the operation of a centrifugal blower.

Utilization of a variable-speed motor to achieve airflow control during the mileage accumulation tests will require a motor almost 50 percent larger. The added power will be required to achieve the higher maximum speed and match the wide open throttle acceleration which occurs during lap 11 of the mileage accumulation test.

Two different types of variable speed drives are commercially available and adaptable for this application. These are the constant-speed AC motor with an eddy current clutch and a DC motor system.

Two manufacturers, Louis Allis and Eaton Dynamatic, have supplied literature describing similar eddy current systems and a typical drive is shown in Figure 5-16. In this drive a standard three-phase induction motor drives the input member of the eddy current clutch at a constant speed.

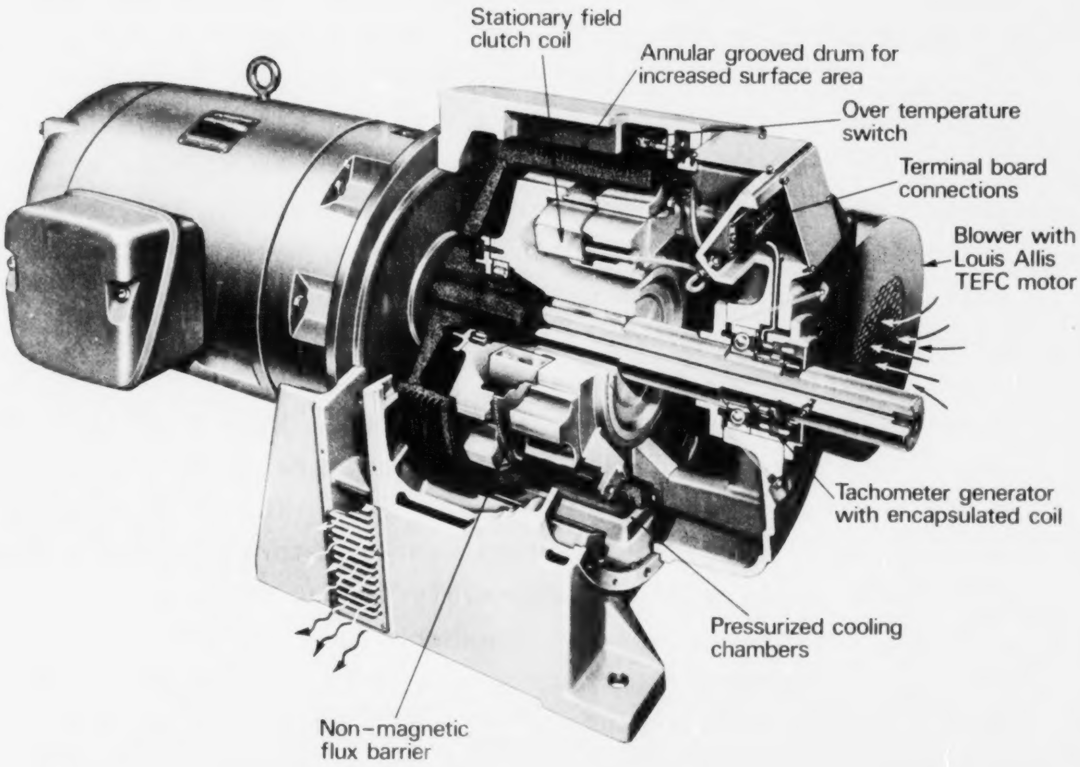

Stationary field clutch coil
Annular grooved drum for increased surface area
Over temperature switch
Terminal board connections
Blower with Louis Allis TEFC motor
Tachometer generator with encapsulated coil
Pressurized cooling chambers
Non-magnetic flux barrier

Figure 5-16. EDDY CURRENT DRIVE

The stationary field coils in the clutch generate magnetic fields which, in turn, generate eddy currents in the input member drum. These eddy currents generate magentic fields which expose the primary fields and give rise to an electromotive force between the AC motor and the load. Speed control is maintained by modulating the DC excitation current to the clutch coil. A basic drive control loop is shown in Figure 5-17. The electronic control must include current feedback curcuits for stability, current limiting circuits for coil protection, regulated power supplied, and isolation transformers for safety, reliability and noise immunity.

A 20 hp drive would be required to operate a vaneaxial blower required for this application, and this drive could deliver sufficient torque to meeet the acceleration demand of the LA-4 cycle, but not necessarily the WOT conditions of the eleventh lap of the mileage accumulation cycle. During the deceleration modes of the LA-4 cycle, supplementary braking would be required. With this type of drive, 0.5 percent requlation can easily be maintained over a 35-to-1 speed range. The approximate price of an Eaton unit, Model 324T is $6,500 with supplementary braking. The Louis Allis equipment is comparably priced.

There are several drawbacks to this system. First the dynamic range of this system is only 35-to-1 as compared to the specified range of 100-to-1. However, this is not a serious deficiency since it is doubtful that a motorcycle being cooled by a 3 km/hr breeze would exhibit significantly different emissions than one cooled by a 1 km/hr wind. A more serious drawback is the response time of the system estimated to be between 0.5 and 1.5 seconds and the power needs of the drive. Since the induction motor is always operating at a constant speed, it will usually be drawing more power than is required to operate the blower. This unused power will be dissipated into the room in the form of heat, increasing the load on the room's heating and air conditioning system.

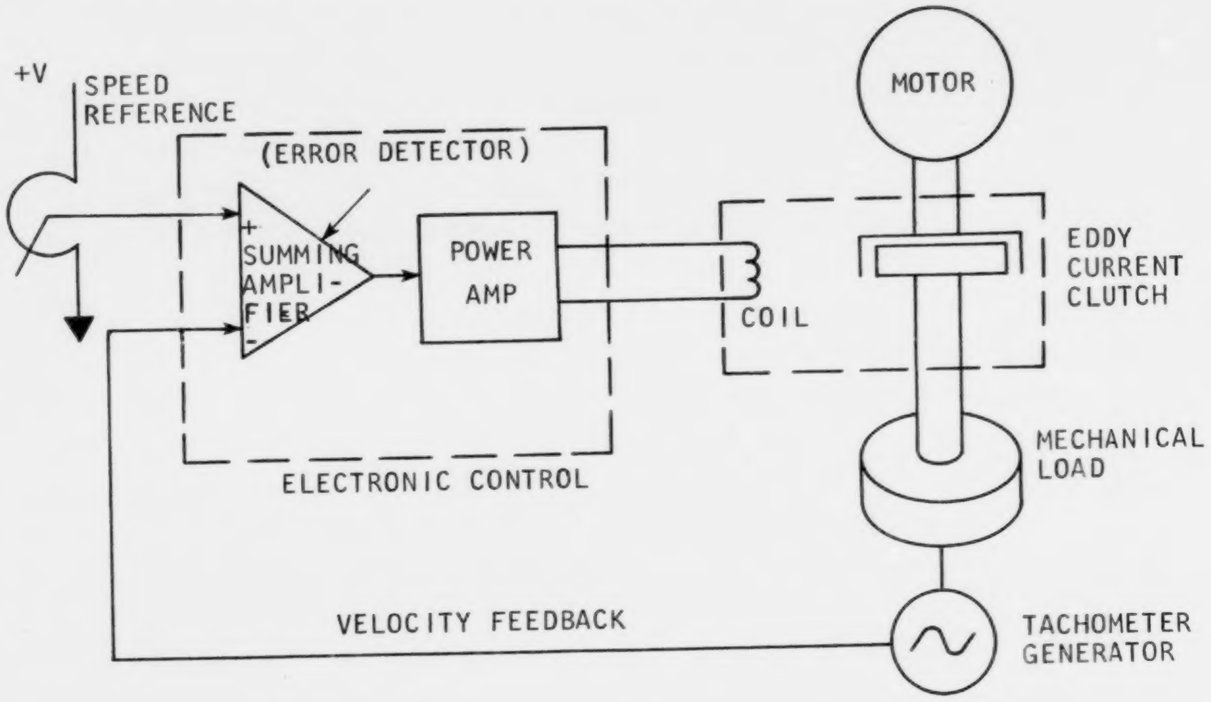

+V
SPEED REFERENCE
(ERROR DETECTOR)
+
SUMMING AMPLIFIER
-
POWER AMP
ELECTRONIC CONTROL
COIL
MOTOR
EDDY CURRENT CLUTCH
MECHANICAL LOAD
TACHOMETER GENERATOR
VELOCITY FEEDBACK

Figure 5-17. BASIC VELOCITY CONTROL LOOP
EDDY CURRENT DRIVE

In spite of these drawbacks, the eddy current unit is well suited for this application. It is a proven, turnkey system which has been utilized in similar applications. While its cost is greater than the bypass approach, it has already proven itself and can be quickly and easily adapted for this application with almost guaranteed satisfaction.

Two types of variable-speed DC drives are available. The first is equipped with a SCR/thyristor controller and the other style employs a regenerative type controller. The latter unit has a wider range of operation, including braking and greater regulation than the former, but there is a cost penalty. Compartive performance data is summarized in Table 5-5.

A representative SCR system, the Max Pak V*S Drive is produced by Reliance Electric Company, and consists of three major components.

- A variable-speed DC motor designed for operation from a phase controller rectified power supply.
- A solid-state power converter and regulator for providing rectified power.
- An operator's control station, which contains the master speed control and the pushbutton control devices to operate the driver.

A typical interconnection diagram is shown in Figure 5-18.

The control unit performs three separate tasks. Its primary function is to rectify the input AC current and regulate the DC output. This is achieved using SCR/thyristor circuitry. The second function of the controller is to provide protection from AC surges, instabilities, etc. The final element is the operator's control panel which contains

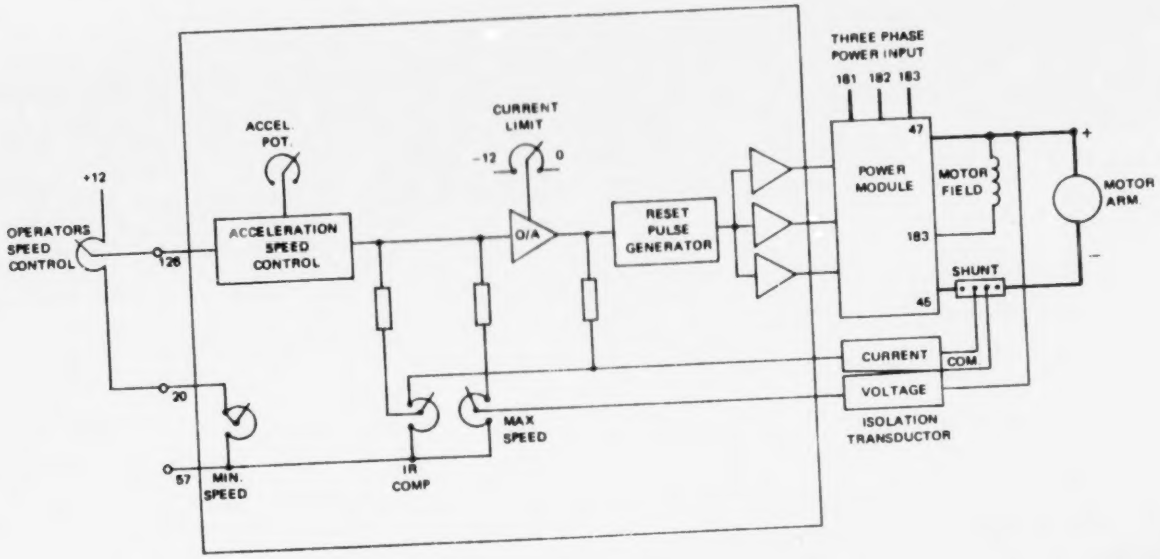

ACCEL. POT.
CURRENT LIMIT
−12
0
+12
OPERATORS SPEED CONTROL
126
ACCELERATION SPEED CONTROL
O/A
RESET PULSE GENERATOR
THREE PHASE POWER INPUT
181
182
183
47
POWER MODULE
MOTOR FIELD
183
45
+
MOTOR ARM.
−
SHUNT
CURRENT
COM
VOLTAGE
ISOLATION TRANSDUCTOR
20
57
MIN. SPEED
IR COMP
MAX SPEED

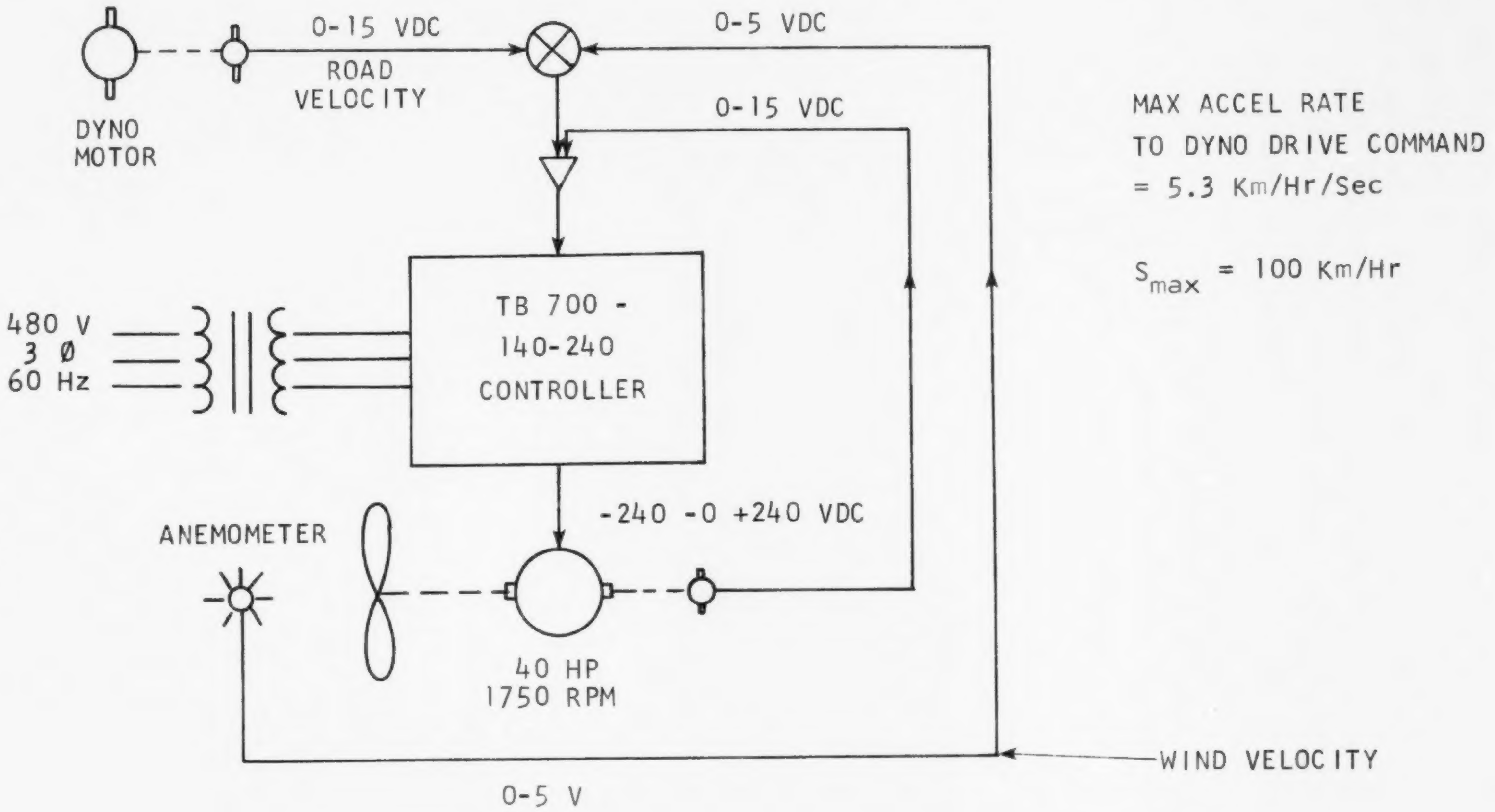
Figure 5-18. TYPICAL INTERCONNECTION DIAGRAM

the master speed control potentiometer, jog-run selector switch, and start-stop pushbuttons.

The variable-speed DC motor has been designed to operate from rectified power and can be modified to include gearmotors, tachometers, slide bases, etc. While motors can be supplied in a variety of base speeds, the most common speed is 1,750 rpm. A substantial cost premium, approximately $1,000 must be paid if a slower speed motor is required. Typically, the drive can be operated at speeds as low as 1/20th of the base speed without providing supplementary cooling for the motor. However, this cooling is mandatory when the motor speed is less than 5 percent of the base speed for sustained periods.

Typically, DC drive systems can provide regulated speed control within ±2 percent over an operating range of 50-to-1. However, optional circuitry can be acquired to improve the regulation to ±0.1 percent and increase the range to 100-to-1. As in the eddy current system, the DC drive can easily meet the acceleration demands imposed by the LA-4 cycle, but no braking is provided to meet the deceleration demands.

The regenerative system is basically the same as the previously described unit. However, additional controlled rectifiers have been added to accomplish regeneration and reversible operation. As a result of the added circuitry, the regenerative system has a rapid response time, less than 500 ms, a wide dynamic range of operation, and does not require supplementary braking. A flow control system which utilizes a Randtronic's regenerative unit is schematically shown in Figure 5-19. This system is somewhat different in that it employs a dual feedback arrangement, using both a tachometer mounted on the motor shaft, and a air velocity sensor. The price of this system is $8,300 and delivery would be 14 weeks after the receipt of an order.


DYNO
MOTOR
0-15 VDC
ROAD
VELOCITY
0-5 VDC
0-15 VDC
480 V
3 Ø
60 Hz
TB 700 -
140-240
CONTROLLER
-240 -0 +240 VDC
ANEMOMETER
40 HP
1750 RPM
WIND VELOCITY
0-5 V

MAX ACCEL RATE
TO DYNO DRIVE COMMAND
= 5.3 Km/Hr/Sec

S_{max} = 100 Km/Hr

Figure 5-19. DC REGENERATIVE CONTROL SYSTEM

5.6.2.4 Compound Control System

A compound control system would employ two or more of the control methods described above. Therefore, it would be more complex than any system employing a single control mechanism. As a result of its more complex nature, the development of a compound system should not be pursued unless no single control mechanism is capable of meeting requirements outlined in Table 5-1.

Since a single control method appears to meet the design objectives of this application, further examination of the more complex compound system is unwarranted at this time.

5.7 RECOMMENDATIONS

5.7.1 Ranking of Alternatives - Blowers

Optimum Selection

1. Centrifugal - Airfoil Impeller
2. Centrifugal Blower - Backward Inclined Impeller

Acceptable Selection

3. Vaneaxial Fans

Unacceptable Selection

4. Mixed Axial Centrifugal Blower
5. Centrifugal Blower - Radial Impeller

Not Applicable

6. Tubeaxial Fans
7. Centrifugal Blower - Forward Curved Impeller
8. Propeller Fans

As a result of the investigation conducted to date, no discernible differences between centrifugal blowers equipped with airfoil blades and backward curved blades could be identified, and it is felt that these units are directly interchangeable. Further consideration should be given to a vaneaxial unit because of its more compact design, low weight, low rotational inertia, uniform outlet air distribution and low cost.

The mixed axial unit and the centrifugal blower equipped with radial blades were considered marginally acceptable because of their high cost, large size, poor efficiency and high noise emission. The other blowers were disregarded because of their inability to satisfy the requirements outlined in Table 5-1.

5.7.2 Ranking of Alternatives Variable Flow Control Methods

Acceptable

1. Eddy Current Clutch with AC Induction Motor
2. Bypass Control - Dual Bypass Ducts
3. DC Regenerative Drive
4. SCR DC Drive

Marginally Acceptable

5. Bypass Control - Single Bypass Duct

Unacceptable

6. Inlet Dampers
7. Outlet Dampers

Of the four acceptable choices, the eddy current drive is recommended. While its cost is somewhat greater than the bypass control alternative, it is an off-the-shelf system which is commonly used in similar applications. While the eddy current drive does not have the range and degree of regulation of the two DC drives, it is felt that its performance is sufficient to satisfy the system objectives, although it may not fully meet the design specifications set out in the discussion. The regenerative drive is recommended over the standard SCR system because of its increased regulation, range, response time and braking capabilities.

The dual duct bypass control is definitely superior to the single duct unit, even though it is somewhat costlier and more complex.

While shut-off dampers provide the least expensive alternative, their dynamic range is insufficient to fulfill the requirements of the project.

5.7.3 System Recommendations

It is recommended that two design alternatives be developed further in Task 6 and that a specification based on the most promising of these designs be written.

The first design would employ a vaneaxial fan and a variable speed drive, probably an eddy current unit, but possibly a regenerative DC drive. The vaneaxial blower was selected for use with this drive because its inherent performance/design characteristics; i.e., low rotational inertia mass; maximum rotation speed of 1,750 RPM; output proportional with rotational speed; match the needs of the drive. The increased costs of the variable-speed drive are offset by the lower costs of the vaneaxial blower. Noise emissions from this system should not be offensive, since the blower will only be operating at maximum flow conditions for a short period. Supplementary cooling of the drive will not be required because the motor will be directly coupled to the fan wheel and will be continually cooled by the inlet air passing over it.

The second system is a centrifugal blower mounted in dual-duct bypass configuration. The centrifugal blower is well suited for this mode of operation because of its high efficiency, low noise output, ability to overcome substantial (5 to 8" H_2O) pressure drop, and high rotational inertia which provides stable operation over prolonged periods of time.

Section 6

TASK 6 - COOLING SYSTEM ANALYSIS

6.1 INTRODUCTION

The design of a variable-speed blower system which simulates road-cooling conditions will be detailed in this report. The requirements as set forth in the EPA Draft Regulations and the contract will be reviewed, and the recommendations of the Task 5 Report, as modified by the EPA Project Officer, will be summarized. Additionally, in this report a cost-trade-off analysis of the design alternatives will be conducted.

6.2 REVIEW OF REQUIREMENTS

Paragraph 85.478-15 (b) of EPA's Notice of Proposed Rulemaking (NPRM) for New Motorcycles, specifies the incorporation of a variable-speed blower system for simulating engine cooling. The Regulations and the contract require that the blower have a mechanism, controlled by the dynamometer roll speed, which will regulate the blower flow to within 10 percent of the roll speed over an operating range of 10 kph (6.2 mph) to 100 kph (62 mph). At speeds less than 10 kph the air flow shall be within ±1 kph.

The NPRM and the contract also describe the configuration and positioning of the blower outlet duct. As stated in the contract, "the cooling system shall be located to discharge air to the front of the motorcycle so that the movement of air over the motorcycle engine simulates the movement of the motorcycle through the air. The cooling system shall have a square or rectangular outlet of at least $0.5m^2$ (5.38 $feet^2$) outlet area. The lower edge of the outlet shall be located about 150mm (5.91 inches) above dynamometer floor level. Outlet flow shall be uniform to ±20 percent across the outlet area as measured at the center of the area as compared to the center of each quarter area."

In the presentation which follows, the rationale behind these requirements will be reviewed, and the feasibility and/or desirability of achieving these objectives will be examined.

6.3 REVIEW OF TASK 5 REPORT RECOMMENDATIONS

In the Task 5 Report, separate rankings for blower types and flow control mechanisms were presented. The acceptable alternatives are listed in Table 6-1. Also in that task report two alternative systems were recommended for further development. The first design utilized a vaneaxial fan with a variable-speed drive such as an eddy current, DC, or regenerative DC motor. The second system employed a centrifugal blower mounted in a dual-duct bypass configuration.

These findings were discussed in detail at a review meeting held with the Project Officer and other members of the EPA staff on May 21, 1975. On May 29, 1975 the Project Officer directed Olson Laboratories to examine only the following two alternatives in the Task 6 Report:

1. A vaneaxial blower used in conjunction with a variable-speed drive.

2. A centrifugal blower used in conjunction with a variable-speed drive.

Table 6-1. TASK REPORT 5 - RECOMMENDATIONS

A. Blowers

Optimum Selection

1. Centrifugal Blower - Airfoil Impeller
2. Centrifugal Blower - Backward Inclined Impeller

Acceptable Selection

3. Vaneaxial Fans

B. Flow Control Methods

Acceptable

1. Eddy Current Clutch with A-C Induction Motor
2. Bypass Control - Dual Bypass Ducts
3. DC Regenerative Drive
4. SCR DC Drive

Marginally Acceptable

1. Bypass Control - Single Bypass Duct

Olson Laboratories concurs that these two systems are viable alternatives, which would result, with a minimum design effort, in systems which achieve the design goals outlined in Section 6.2.

6.4 DESIGN ALTERNATIVES

As stated in Section 6.3, the Project Officer has requested that two design alternatives be examined in detail. As a result of this study, a final design will be developed and specifications will be formulated. In this section of the task report, evaluation criteria will be defined and a cost-effectiveness study of the design alternatives will be conducted. The alternatives will be ranked and a recommendation made.

In reality there are six system designs which should be considered. These combine the two types of blowers with the three types of variable-speed drives. The six combinations are:

1. Vaneaxial blower with eddy current drive.
2. Vaneaxial blower with DC drive.
3. Vaneaxial with regenerative DC drive.
4. Centrifugal blower with eddy current drive.
5. Centrifugal blower with DC drive.
6. Centrifugal blower with regenerative DC drive.

Additionally, there are several suppliers for each component. Characteristics of representative units have been utilized in this analysis. However, specific design features which either greatly improve or degrade a component's performance in this application will be noted, and the specification will appropriately be revised. In no case will the specification be so limited that only one supplier can meet the requirements.

6.4.1 Ranking System

In order to evaluate the alternative designs, a number of evaluation criteria have been established. These have been divided into two sections; the first discusses criteria related to the blower. and the second addresses the motor control system and the resulting system performance.

All of the alternatives will be evaluated with respect to each of the criteria and will be rated on a scale of 1 to 5, where 1 is unacceptable, 3 is acceptable, and 5 is outstanding. Additionally, a weighting factor has been designated for each of the criteria, defining its relative importance. Each of the ratings will be adjusted by using the weighting factor, and the rankings of the alternatives will be calculated by summing the weighted ratings.

The evaluation criteria with their respective weighting factors are listed in Table 6-2, as well as the unweighted ratings.

6.5 EVALUATION CRITERIA - BLOWER

6.5.1 Flow as a Function of RPM and ΔP

In order to define the blower control system, the relationship between the blower output and the rotational speed of the impeller must be established. Ideally, this relationship should be linear; thus permitting a simple tachometer control circuit. If the relationship is non-linear, then, as reported in the Task 5 Report, a velocity sensor must be used as the feedback sensor in the control system which will result in a more complex control circuit and ducting arrangement.

The fan laws state that

$$\text{new CFM} = \text{old CFM} \times \frac{\text{new rpm}}{\text{old rpm}} \qquad (6\text{-}1)$$

Table 6-2. UNWEIGHTED RATINGS

CRITERION	WT. FACTOR	SYSTEM NO. (SECT. 2.2)					
		VANEAXIAL BLOWER			CENTRIFUGAL BLOWER		
		EC	DC	RDC	EC	DC	RDC
I. Blower Performance							
1. Flow as a function of rpm and ΔP	1.0	5	5	5	2	2	2
2. Rotational Inertia	0.9	3	3	3	2	2	2
3. Delivery	0.8	3	3	3	3	3	3
4. Noise	0.8	3	3	3	4	4	4
5. Reliability/Maintenance	0.8	4	4	4	3	3	3
6. Adaptability	0.8	3	3	3	3	3	3
7. Power Consumption/ Efficiency	0.7	2	3	3	2	4	4
8. Motor Compatibility	0.7	3	2	2	4	4	4
9. Size and Weight	0.6	4	4	4	3	3	3
10. Flow Conditioning	0.5	4	4	4	2	2	2
11. Cost	0.5	4	4	4	3	3	3
II. Motor Control and System Performance							
1. Accuracy	1.0	4	1	4	2	1	2
2. Range	0.9	2	4	4	1	1	1
3. Response Time	0.9	3	1	3	2	1	2
4. Delivery	0.8	3	3	3	3	3	3
5. Size and Weight	0.7	3	3	3	3	3	3
6. Adaptability	0.7	4	1	4	2	1	2
7. Reliability/Maintenance	0.7	4	3	3	4	3	3
8. Interferences	0.7	3	3	3	3	3	3
9. Cost	0.7	3	3	3	3	3	3

and

$$\text{new static pressure (SP)} = \text{old SP} \times \left(\frac{\text{new rpm}}{\text{old rpm}}\right)^2 \quad (6\text{-}2)$$

However, these formulas are most accurate when used to define conditions at two operating points which are fairly close together. Under the conditions imposed by this application, the blower will be operating over a 100-to-1 range, and over this range, the validity of the fan laws is marginal.

In order to establish flow versus rpm curves for vaneaxial and centrifugal blowers of the appropriate capacity, the static pressure requirements for this application, as a function of flow, must be defined. The static pressure which the blower must overcome consists of two components. The first occurs as frictional losses in the ducting and is a function of the ducting configuration, ducting smoothness, and mean air velocity. The second component results from pressure losses occurring as a result of contractions and expansions in the duct cross-section. The magnitude of this loss is affected by the ratio of the two duct areas and the volumetric flow of air.

A family of curves has been plotted in Figure 6-1 defining the total static pressure loss as a function of blower exit area for several volumetric flow rates. For the purposes of those calculations, the duct exit area has been assumed to be $0.5m^2$ (5.38 feet^2).

Utilizing performance data supplied by manufacturers and the static pressure demands defined in Figure 6-1, performance curves relating air delivery and rotational speed were developed for four blowers, two vaneaxial units and two centrifugal units. These curves are shown in Figure 6-2. Performance data, supplied by the manufacturer, describe the operation of a blower in a typical industrial application, in which the blower operates at a specified condition or over a confined range of conditions, usually no more than a 10-to-1 range. In this application, the blower will be required to operate over a 100-to-1 range. Therefore, in order to define operation at slower speeds, the manufacturer's data has been extrapolated.

In the case of the centrifugal units, this extrapolation has provided some valuable information. Referring to Figure 6-2, the larger centrifugal unit displayed a linear relationship between rotational speed and air flow over a 3-to-1 range of operation, while the smaller unit appeared to be linear over a 2-1/2-to-1 range. When these performance curves were extrapolated down to a no-flow condition, however, a negative rotational speed resulted. This, of course, does not occur in the real world. Therefore, since the performance curve must go through the origin of this plot, it is apparent that the performance curve of the centrifugal blower is not linear over the entire range of interest. In contrast, the vaneaxial fans performance curves are linear and do appear to go through the origin when the performance curve is extrapolated to a zero-flow condition.

Based on these curves, the vaneaxial blower appears to have a distinct advantage over the centrifugal unit when either an open-loop or closed-loop tachometer control system is used.

6.5.2 Fan Wheel Rotational Inertia

The inertia of the blower's driven element at the motor shaft is an important parameter affecting the selection of the motor and control system. The motor must exert an accelerating torque sufficient to overcome the fan wheel's inertia and accelerate it to the operating speed. The acceleration rate is a function of the magnitude of the inertia.

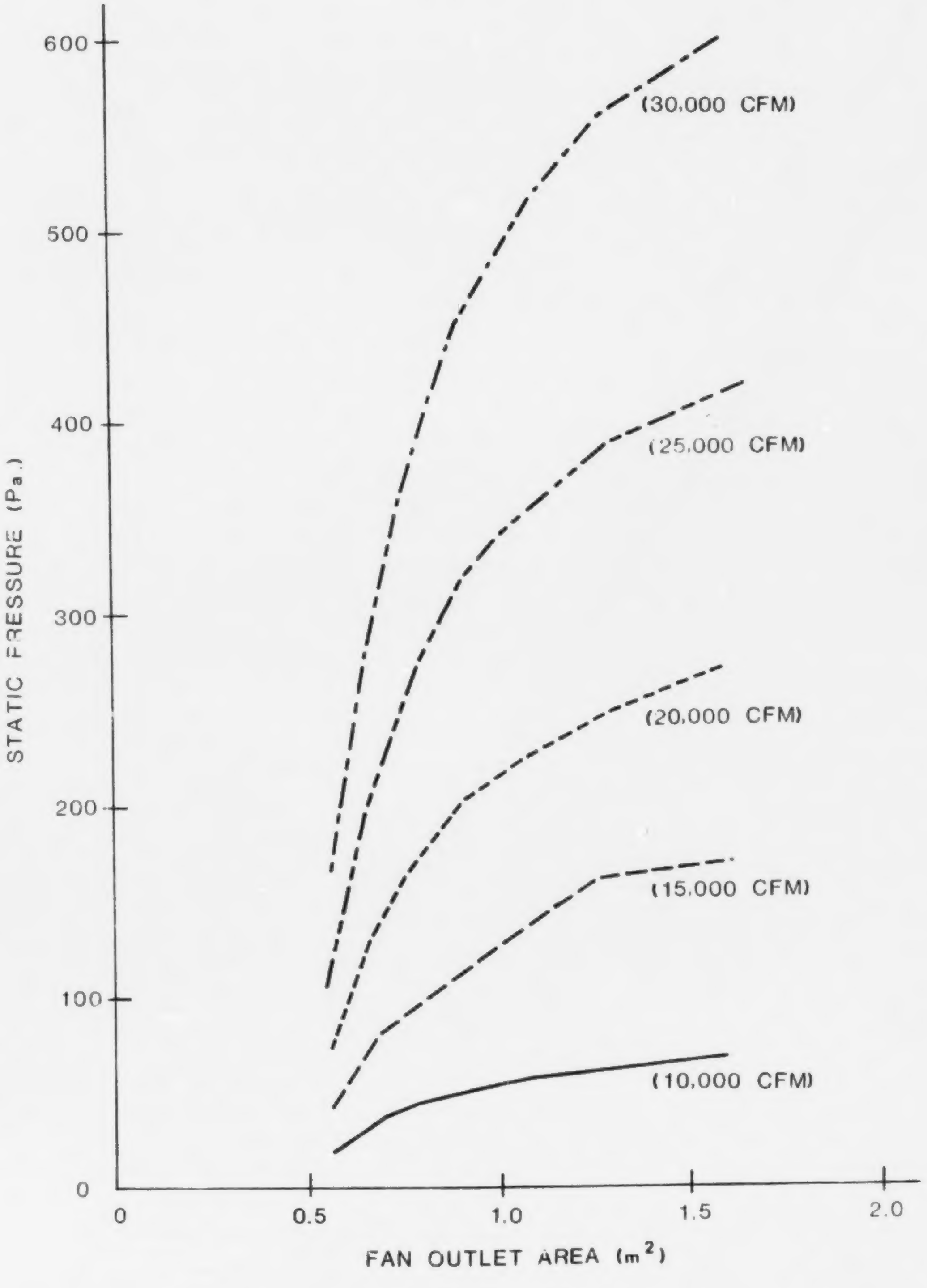
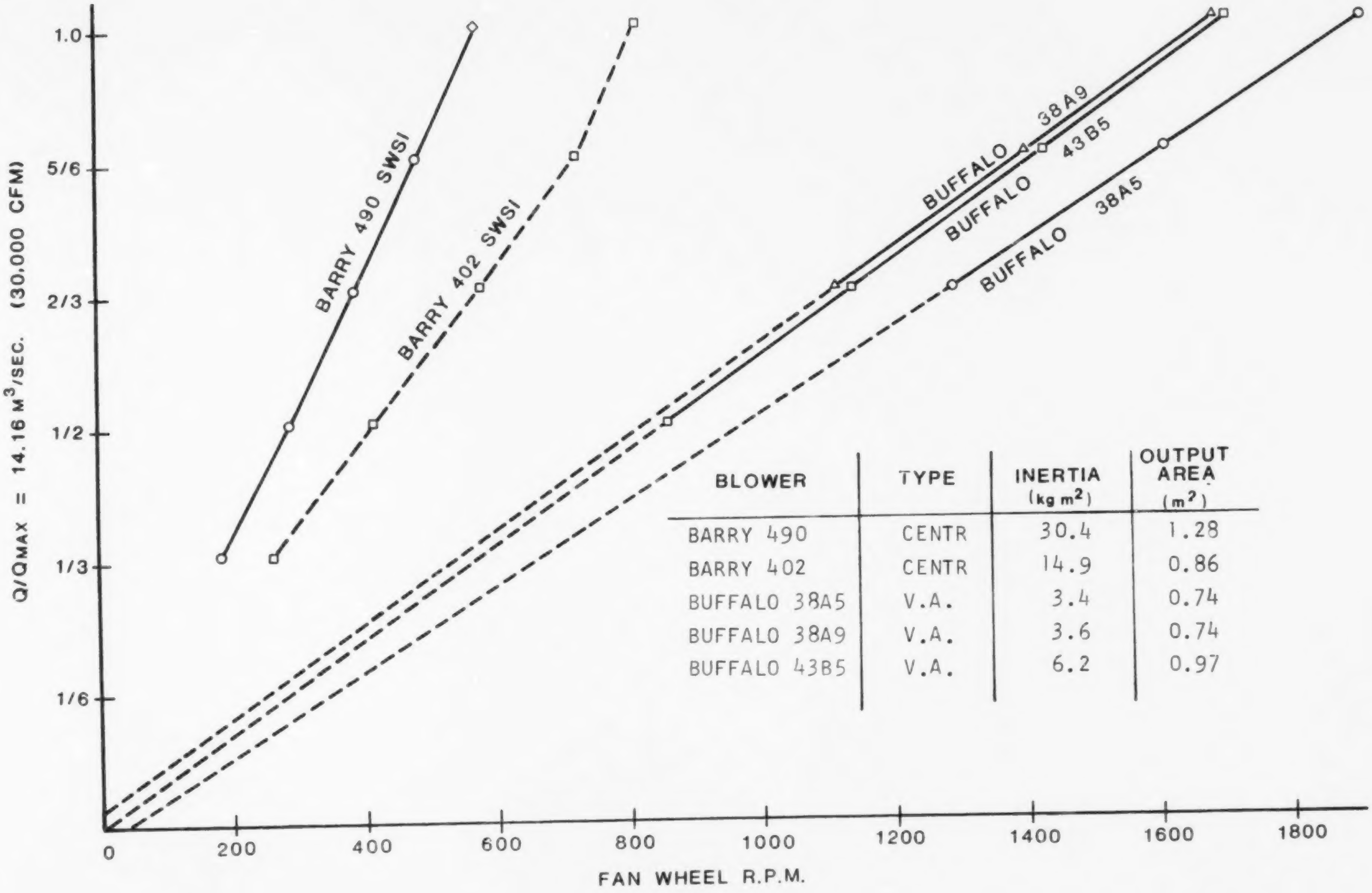


FIGURE 6-1. STATIC PRESSURE VS. FAN OUTLET AREA



BLOWER	TYPE	INERTIA (kg m^2)	OUTPUT AREA (m^2)
BARRY 490	CENTR	30.4	1.28
BARRY 402	CENTR	14.9	0.86
BUFFALO 38A5	V.A.	3.4	0.74
BUFFALO 38A9	V.A.	3.6	0.74
BUFFALO 43B5	V.A.	6.2	0.97

FIGURE 6-2. FAN FLOW VS. ROTATIONAL SPEED

Data on acceleration rates as a function of fan wheel inertia were obtained for both eddy current and DC motors. This data is summarized in Table 6-3.

Table 6-3. FAN WHEEL ACCELERATION RATES
(Zero to Max RPM)

BLOWER	MAX RPM	INERTIA ($kg\ m^2$)	EC MOTOR ACCEL[1] (sec)	DC MOTOR ACCEL[2] (sec)
Hartzell VA 36A	1,750	2,73	1.9	2
Barry 445 SISW	800	21.30	3.1	5
New York Blower 449 SISW	890	25.40	3.6	8

[1] Data supplied by Louis Allis Co. 40 hp Adjusto-Speed Motor.

[2] Data supplied by Robicon Corp. 40 hp General Electric Motor.

As shown in the table, the vaneaxial Hartzell blower can be accelerated to its maximum speed faster than the centrifugal units even though its maximum rotational speed is twice that of the centrifugal units. This demonstrates the importance of fan wheel inertia.

As shown in Table 6-3, there appears to be a difference in the acceleration characteristics of the two types of motor controls. However, motor vendors indicate that acceleration rates are dependent on the motor design, specifically the armature inertia. Therefore, for each class of motors, various acceleration rates are available. Achievable acceleration rates are greater than those required to meet the objectives of this program. In the specification, a minimum acceleration rate and response time are imposed by the accuracy specification.

Deceleration rates will also be dependent on the inertia of the fan wheel. However, deceleration times without the use of external braking or regenerative techniques will be long, probably in excess of a minute, and unacceptable for this application. Braking requirements will be discussed in Section 6.6.3. .

6.5.3 Delivery Schedule

Time required for delivery of the system is considered important because it could impact on the EPA's plans to institute a certification program on 1978 motorcycles. Several manufacturers have been contacted to determine representative delivery schedules. These are summarized in Table 6-4. As shown in this table delivery dates are reasonable, and there appears to be no distinct differences between deliveries of vaneaxial and centrifugal units.

Table 6-4. TYPICAL DELIVERY INFORMATION

MANUFACTURER	MODEL NO.	ARRANGEMENT NO.	DELIVERY (days)ARO	$	SHIPPING WT. (kg)
Barry Blower	402	7 SWSI	90	1509	544
	455	7 SWSI	90	1830	680
	490	7 SWSI	90	2171	771
New York Blower	449	8 SWSI	90	2360	742
	449	8 SWSI	90	2964	952
Aerovent	V361	Type W	70	1041	272
	V381	Type W	70	1219	327
Buffalo Forge	33A5	4 Type S	125	875	142
	33A9	4 Type S	125	1058	142
	38A5	4 Type S	125	956	166

6.5.4 Noise

Noise levels within the test cell must be kept at tolerable levels. As discussed in the Task 5 Report, expected noise levels from a vaneaxial unit will be greater than those for a comparable centrifugal unit. A comparison of the expected noise levels is shown in Table 6-5. Sound

Table 6-5. NOISE LEVELS

BLOWER	TYPE	SOUND POWER LEVELS dB re 10^{-12} WATT								dBA[1]
		20 - 75 Hz	75 - 150 Hz	150 - 300 Hz	300 - 600 Hz	600 - 1200 Hz	1200 - 2400 Hz	2400 - 4800 Hz	4800 - 10000 Hz	
		@ 835 m^3/min. (29,500 cfm)								
New York Blower 449	Centr.	98	101	91	89	87	80	74	70	90
New York Blower 490	Centr.	103	96	91	85	84	81	76	74	84
Aerovent V361-Y34	V.A.	101	94	95	98	98	96	88	82	88
Aerovent V381-Y30	V.A.	102	94	95	99	98	95	89	82	87
		@ 417 m^3/min. (14,750 cfm)								
New York Blower 449	Centr.	83	86	76	74	72	65	59	55	75
New York Blower 490	Centr.	88	81	76	70	69	66	61	59	69
Aerovent V361-Y34	V.A.	86	79	80	83	83	81	73	67	73
Aerovent V381-Y30	V.A.	87	79	80	84	83	80	74	67	72

[1] dBA estimated at 1.5 m from blower outlet

power levels and dBA levels have been reported in the table for two different flow conditions, maximum flow and one-half maximum flow.

The dBA noise levels reported in Table 6-5 do not exceed OSHA standards which permit an individual to be exposed to sound levels 90 dBA and below for periods longer than 8 hours. Some care must be taken by the facility designer, however, to protect test-cell areas from the high frequency noise emitted by the vaneaxial unit.

The estimated sound magnitudes do not warrant the installation of sound attenuators. While attenuators would reduce noise levels significantly, they introduce added static pressure losses which would increase horsepower requirements and costs. Maximum sound levels of 90 dBA and average sound levels of 75 dBA are tolerable, and well within acceptable limits.

6.5.5 Reliability/Maintenance

The reliability and maintenance requirements for the two blower types were examined. Both units are highly reliable because of their inherent simplicity.

The vaneaxial blower, excluding the motor, requires no maintenance since the motor is connected directly to the impeller wheel and the blower itself has no bearings. This is in contrast to the centrifugal blower which utilizes two bearings, one on each side of the impeller to support the impeller wheel. These bearings are typically grease-lubricated, heavy-duty and self-aligning. The average service life of these bearings is 125,000 hours using B-10 minimum-life-rating data as defined by the Anti-Friction Bearing Manufacturers Association. The minimal servicing required for the centrifugal blowers should not preclude their use in this application.

Table 6-5. NOISE LEVELS

BLOWER	TYPE	SOUND POWER LEVELS dB re 10^{-12} WATT								dBA[1]
		20 - 75 Hz	75 - 150 Hz	150 - 300 Hz	300 - 600 Hz	600 - 1200 Hz	1200 - 2400 Hz	2400 - 4800 Hz	4800 - 10000 Hz	
		@ 835 m^3/min. (29,500 cfm)								
New York Blower 449	Centr.	98	101	91	89	87	80	74	70	90
New York Blower 490	Centr.	103	96	91	85	84	81	76	74	84
Aerovent V361-Y34	V.A.	101	94	95	98	98	96	88	82	88
Aerovent V381-Y30	V.A.	102	94	95	99	98	95	89	82	87
		@ 417 m^3/min. (14,750 cfm)								
New York Blower 449	Centr.	83	86	76	74	72	65	59	55	75
New York Blower 490	Centr.	88	81	76	70	69	66	61	59	69
Aerovent V361-Y34	V.A.	86	79	80	83	83	81	73	67	73
Aerovent V381-Y30	V.A.	87	79	80	84	83	80	74	67	72

[1] dBA estimated at 1.5 m from blower outlet

6.5.6 Adaptability

The system which is being specified in this report has been designed around the requirements imposed by the Federal Test Procedure (FTP), but it also should be flexible enough to satisfy other potential requirements. These added requirements could require increased flow capacity which can be achieved easily with either style of blower. However, all techniques for additional flow require more power.

Added flow output from a centrifugal blower can be achieved by increasing the rotational speed of the impeller. This is easily accomplished because motors which are readily available for this application have a maximum operating speed of 1,170 or 1,750 rpm, which is substantially faster than the 600 to 800 rpm needed to satisfy the basic requirements imposed by the FTP. Before selecting a centrifugal blower, the compatability of the blower's structural design and the maximum rotational speed required should be verified, since all blowers have maximum operating speeds. As an example, Table 6-6 defines the maximum safe rpm of the New York Blower units at ambient conditions.

Table 6-6. MAXIMUM SAFE WHEEL RPM
OF SWSI BLOWERS

SIZE	CLASS I	CLASS II	CLASS III
409	1005	1315	1655
449	910	1190	1495
449	850	1105	1395

Ref: Bulletin 725 B New York Blower Company

The output from a vaneaxial blower can also be increased by operating the impeller wheel at faster rotational speeds. Typically, however, these units are usually operated and rated at a rotational speed of 1,750 rpm which is the rated maximum speed of standard variable-speed motors. Increased flow from vaneaxial units can also be achieved by changing the pitch of the blower vanes. This effect can be seen in Figure 6-3 which describes the performance of a vaneaxial blower manufactured by Buffalo Forge. In this style blower, operating at 1,750 rpm, a 280 percent increase in flow can be achieved by adjusting the pitch of the blades. Almost all manufacturers of vaneaxial units have adjustable pitch wheels which permit the user to set manually the pitch of the blades. Two manufacturers supply units in which the pitch can be adjusted pneumatically or electrically while the fan is in motion. However, these units do not have sufficient dynamic range to meet the 100-to-1 variations required for this application.

It is concluded that both blowers can be easily adjusted for increased flow operation if the motors and blower are carefully selected.

6.5.7 Power Consumption/Efficiency

The mechanical efficiency (ME) of a fan is defined by equation (6-3).

$$ME = \frac{Q \, P_T}{672 \, W} \qquad (6\text{-}3)$$

where Q = volumetric flow from blower (m^3/min)
P_T = total pressure (Pa)
W = shaft power delivered by motor (KW)

The average mechanical efficiency of the blower system during one Federal Test Cycle is:

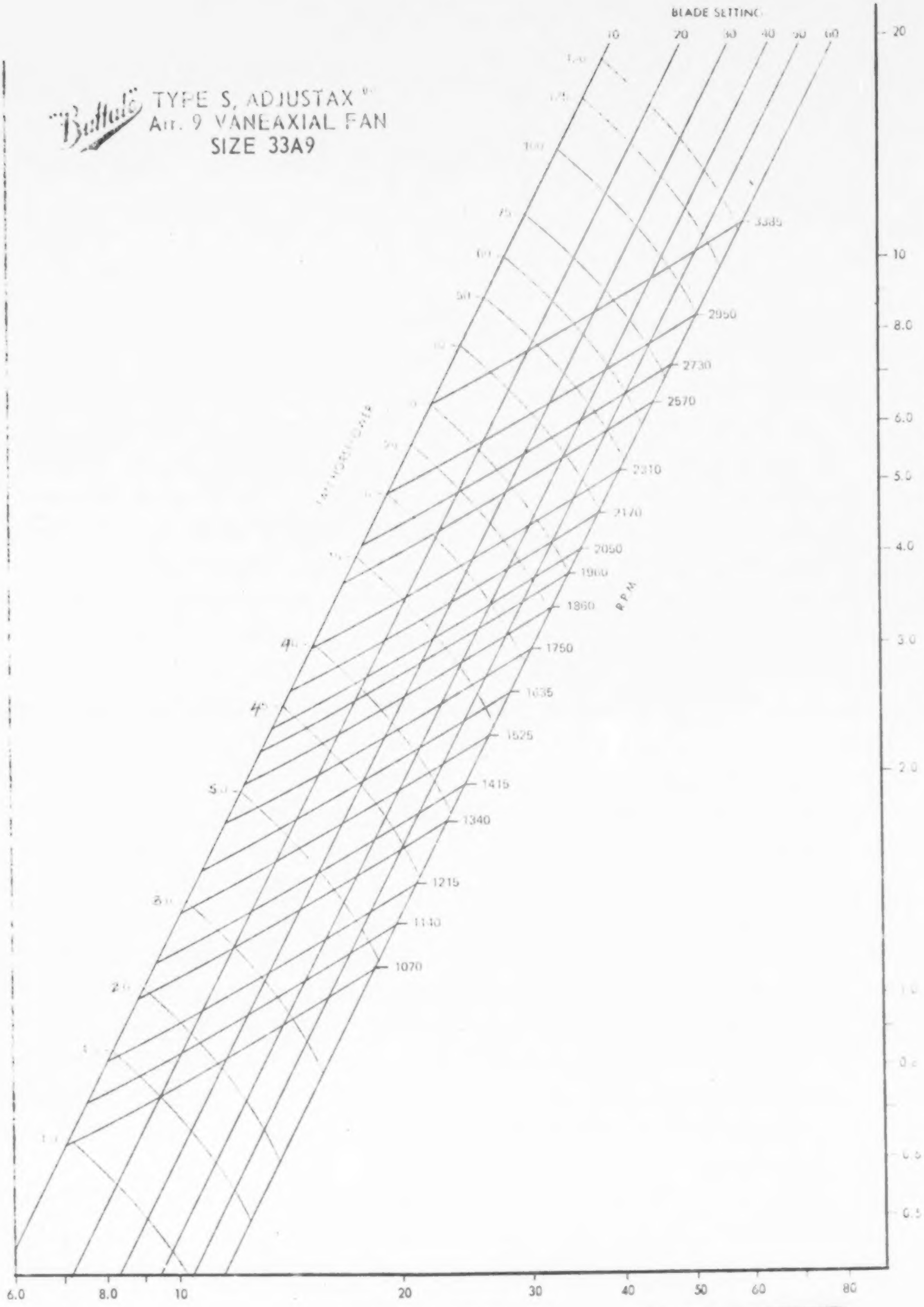


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FIGURE 6-3. VANEAXIAL BLOWER PERFORMANCE

$$ME = \frac{1}{t} \int_{0}^{1371} \frac{Q(t)\, P_T\,(t)}{672\, W(t)} dt \qquad (6\text{-}4)$$

As shown in equation (6-4), the volumetric flow, total pressure and shaft power are all functions of time; varying as the motorcycle speed varies. Also the total pressure and shaft power functions will be different for each blower under consideration. Establishing the efficiency of the total system is further complicated because the efficiency of the drive motor should also be considered, and that varies with shaft rpm. Also, the function W(t) cannot be defined due to lack of suitable data. Blower manufacturers have not developed the necessary data because operation of a blower at 1 percent of its maximum speed, is a condition which is never encountered by the average user.

Therefore, rather than explicitly calculating equation (6-4) for each blower configuration and size, a semi-quantitative study has been conducted, comparing the power requirements and efficiency of several blowers at different air volume outputs. The results have been plotted in Figures 6-4 and 6-5.

In Figure 6-4, the power requirements of several blowers are examined. On the average, the vaneaxial units require more power than the centrifugal units, and the power required by the vaneaxial unit appears to be independent of the blower size. In contrast, the power required to operate a centrifugal blower appears to decrease as the blower size increases.

The efficiency of the various blowers are compared in Figure 6-5. As shown in the plots, vaneaxial units are

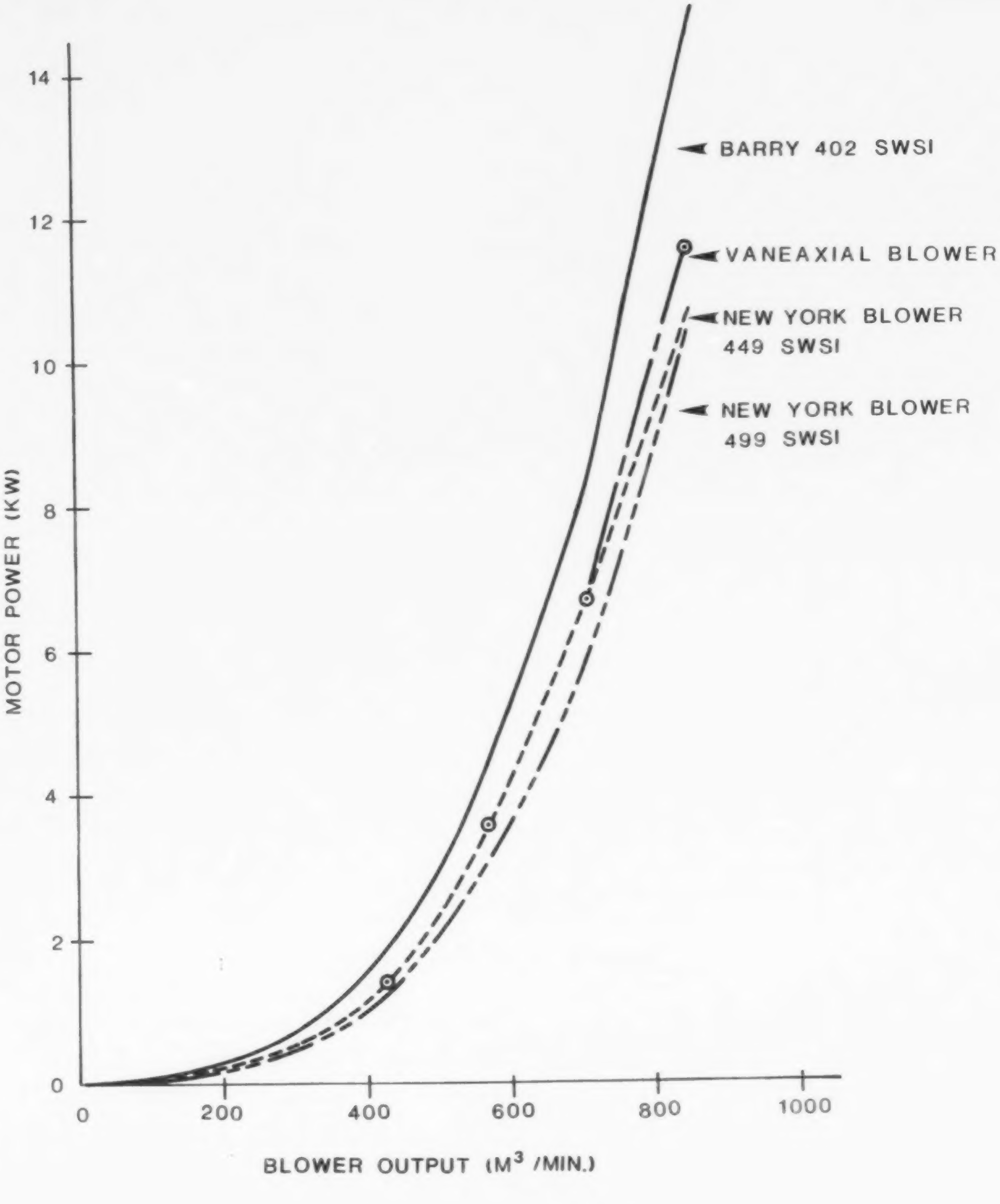
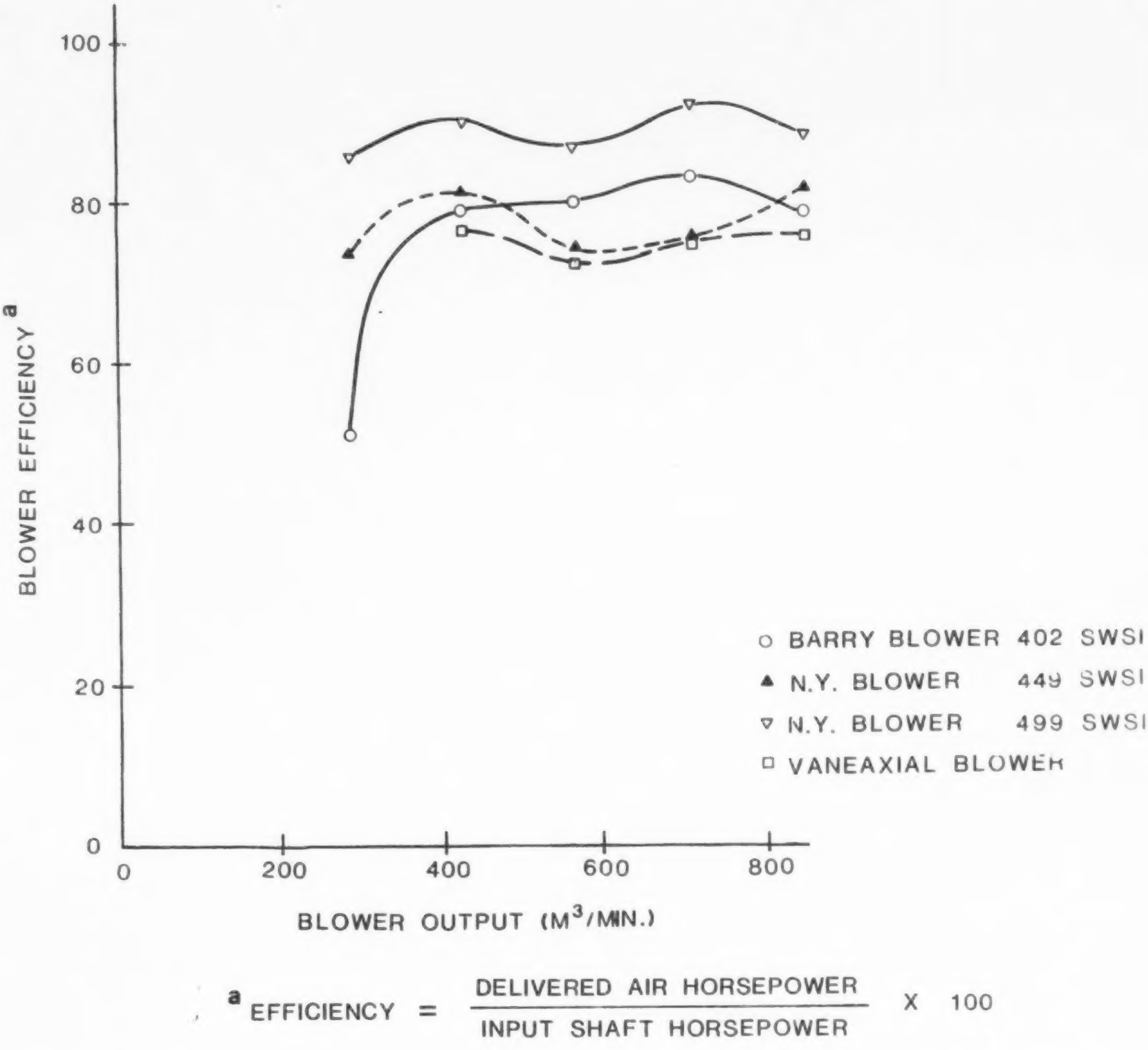


FIGURE 6-4. BLOWER POWER REQUIREMENTS
VS. BLOWER TYPE AND SIZE



$$^{a}\text{EFFICIENCY} = \frac{\text{DELIVERED AIR HORSEPOWER}}{\text{INPUT SHAFT HORSEPOWER}} \times 100$$

FIGURE 6-5. BLOWER EFFICIENCY VS. BLOWER TYPE AND SIZE

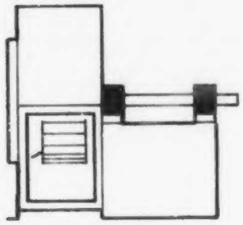
less efficient than comparable centrifugal units.

When considering system efficiency and power consumption, one must evaluate the motor as well as the blower. Efficiency estimates have been obtained for both eddy current and DC motors. The DC motor is not as efficient at maximum speeds as the eddy current unit. The former is 70 percent efficient while the eddy current unit exhibits an efficiency over 90 percent. However, as the rotational speed decreases the situation reverses. The efficiency of the DC motor remains constant over the motor's speed range, but that of an eddy current unit drops to about 40 percent at half speed. At quarter speed, its efficiency is less than 25 percent.

6.5.8 Motor Compatibility

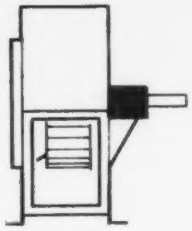
As proposed in the Task 5 Report and reconfirmed in this report, a direct-drive motor should be used to power the blower. This requires that the blower and motor be physically compatible. The use of a direct-drive motor with each of the blower styles under consideration imposes certain design considerations and limitations. These will be subsequently discussed.

Centrifugal blowers are available in several direct-drive configurations. These are shown in Figure 6-6. Arrangements 1, 2, and 3 require the user to fabricate a motor base and interface the motor and the blower. Arrangements 4, 7, and 8 provide a motor base designed by the blower manufacturer to mate with the specific drive being used. The difference between these latter three arrangements is the use and positioning of bearings. Arrangement 4 utilizes no bearings, while two bearings are used in the other two arrangements. For this application, Arrangement 7 is superior. Its use of two bearings, one on each side of the fan wheel, provides excellent support for the unit and maximizes bearing life and utility.



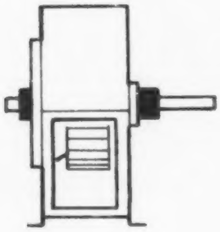
ARRANGEMENT 1

Two bearings on base
Wheel overhung
No motor base



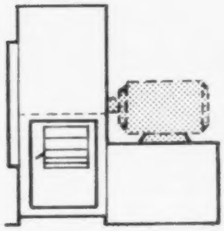
ARRANGEMENT 2

Bearings in bracket
supported by housing
Wheel overhung
No motor base



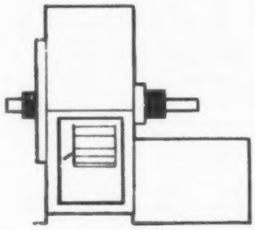
ARRANGEMENT 3

One bearing on each
side of wheel
No motor base



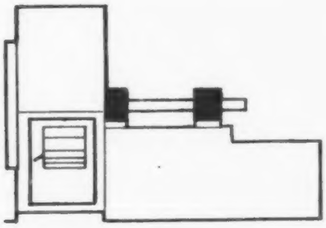
ARRANGEMENT 4

No bearings on fan
Wheel overhung on
motor
Motor base



ARRANGEMENT 7

Same as Arrangement 3
plus motor base



ARRANGEMENT 8

Same as Arrangement 1
plus motor base

FIGURE 6-6. CENTRIFUGAL BLOWER MOTOR MOUNTING CONFIGURATIONS

With the centrifugal blower configurations shown in Figure 6-6, there is no limitation to the size of the motor used. Either drip-proof or totally enclosed motors can be used. The only precaution necessary is that the blower and motor base be securely fastened to the test cell floor.

Motor size is a critical parameter in the design of a vaneaxial fan system. In a direct-drive vaneaxial fan, the motor is close-coupled to the fan wheel and sits within the fan housing. The motor diameter must be smaller than the hub diameter of the fan wheel. Otherwise, the motor will interfere with the movement of air within the fan; thus decreasing its efficiency and capacity. Table 6-7 details the maximum motor frame size which can be accepted by a representative group of vaneaxial fans, and it also defines the frame size of the variable-speed motors being considered. As shown in Table 6-7, the available motors barely fit within the fan housings. If a totally enclosed motor is required, the fit is even tighter and the selection of compatible motors and fans is severely limited.

The compatibility of fan and motor is further affected by the specific design of the fan. As an example, the Buffalo fans have been designed for use only with motors having C-faced flanges (no feet). Although this option is available from the DC motor manufacturers, it is not compatible with eddy current motors where feet are required to reduce the stress at the motor face.

Motor design is another parameter which must be considered. Buffalo Forge only recommends motors that are totally enclosed with air-over construction. This design prevents the free interchange of air between the inside and outside of the motor, resulting in excellent protection to all internal motor components. Hartzell Propeller also recommends the use of totally enclosed motors. However, Aerovent states open-type motors can be used, if they have a

Table 6-7. MOTOR/VANEAXIAL FAN COMPATIBILITY

MOTOR FRAME SIZE

BLOWER MFG	FAN SIZE	MAX MOTOR FRAME SIZE
Aerovent	33	256 T
	36	324 TC
	38	324 TC
Buffalo	33	286 C
	38A	286 C
	43B	365 C

T - Foot mounted
C - Flanged
TC - "C" Flanged Foot mounted

MOTOR SIZES

MFG	MOTOR TYPE	MOTOR RATING (HP)	MOTOR FRAME SIZE	
			DRIP-PROOF	TOTALLY ENCL.
Sabina Electric	DC	20	284 A	324 A
		25	284 A	328 AT
		30	324 A	366 AT
Randtronics and Reliance	DC	20	259 AT	327 AT
		25	287 AT	328 AT
		30	288 AT	366 AT
Wer	EC	20	286 TD	--
		25	324 TSD	--
		30	324 TSD	--
Eaton	EC	20	286 T	286 T
		25	284 T	284 T
		30	286 T	286 T

service factor of 1.15 at 40.5^{o}C (105^{o}F), and noncorrosive air is being passed over the motor.

In conclusion, there are no problems in selecting a direct-drive, variable-speed motor for use with a centrifugal blower. In contrast, variable-speed motors are available for use with vaneaxial blowers, but careful examination of design features must be carried out to ensure that the selected motor and blower are compatible. If a totally enclosed motor is required for use with a vaneaxial blower, an eddy current motor appears to offer some size advantages over a DC unit.

6.5.9 Size and Weight

As discussed in the Task 5 report, there is a substantial difference in the weight and the dimensions of the two blowers being considered for this application. These differences necessitate different blower mounting techniques and ducting designs.

Table 6-4 provided data on the weight of each style blower. As was seen from this data, the mass of the vaneaxial units, as manufactured by Aerovent and Buffalo Forge, is one-fifth to one-half the mass of comparable centrifugal units. These masses do not include the motor.

The mass of eddy current and DC motors, comparably rated, are similar. Typical motor weights are shown in Table 6-8. The motor mass is of the same magnitude of the weight of the vaneaxial units and one-third to one-half the weight of the centrifugal units.

Table 6-8. MOTOR WEIGHTS

MANUFACTURER	MOTOR TYPE	FRAME SIZE	WEIGHT (kg)
Reliance	20 HP DC Drip-Proof	259 AT	188
	25 and 30 HP DC Drip-Proof	287 AT	261
	20 and 25 HP DC Totally Encl.	327 AT	351
	30 HP DC Totally Encl.	366 AT	454
Eaton	20 HP EC Drip-Proof	286 T	252
	25 HP EC Drip-Proof	284 T	252
	30 HP EC Drip-Proof	286 T	454

As a result of the centrifugal unit's size and mass, it must be securely mounted to the test cell floor. Therefore, in order to accomodate various-sized motorcycles, while maintaining a fixed distance between duct exit and the front wheel of the bike, telescoping ducts must be used.

Mounting configurations for the vaneaxial unit are more flexible because of its lighter weight and smaller size. Conversations with vendors have confimred that the vaneaxial system can be mounted on rails and be moved to the appropriate position for testing the various sized units.

6.5.10 Flow Conditioning

The contract specifies that the outlet flow from the blower shall be uniform to ±20 percent across the outlet area as measured at the center of the area as compared to the center of each quarter area. An analysis of the velocity profiles, as it impacts on the ducting design, will be examined in Section 6.7 but consideration should be given at this time to the general characteristics of each style blower and available flow-conditioning accesories.

A propeller or axial flow-type wheel imparts a rotating or screw action to the exit air. In a vaneaxial fan, stationary guide vanes are located directly behind the fan wheel. These vanes reduce air turbulence and partially convert the rotative energy imparted to the air to axial energy. The result is an increase in efficiency, a reduction in noise and nonswirling air flow leaving the fan. In addition, an inlet bell should be installed on the entrance to the vaneaxial unit. This reduces the static pressure loss associated with introducing the air into the fan.

The velocity profile of the air exiting a centrifugal blower is not uniform across the entire cross section of the blower exit. This nonuniformity results from the centrifugal energy imparted to the air by the wheel. The degree of nonuniformity will be a function of the impeller speed with maximum velocity asymmetry occuring at maximum rotational speeds. Flow conditioning techniques, such as turbulators and vane control can be used to condition the exit air. However, the exact flow-conditioning design cannot be developed until more specific data is available.

6.5.11 Costs

Typical blower costs were shown in Table 6-4. As shown, vaneaxial units are approximately 50 percent less costly than comparable centrifugal units. In addition, the costs associated with shipping the vaneaxial units are also less because of its lower weight.

6.6 EVALUATION CRITERIA - MOTOR CONTROL AND SYSTEM PERFORMANCE

6.6.1 Accuracy

As specified in the contract, the linear air velocity from the blower shall be within 10 percent of the motorcycle speed for speeds between 10 and 100 kph and within ±1 kph at speeds below 10 kph. In order to achieve this goal, a variable speed drive equipped with a tachometer follower control or shaft coupling circuitry is recommended. This control enables the driver to follow a signal from the tachometer generator located on the dynamometer roll. The accuracy of the control is dependent on the design of the control circuitry, the characteristics of the tachometers located on the dynamometer roll and fan wheel shaft, and the linearity of the relationship describing blower output versus fan wheel rpm.

As described in Section 6.5.1, the vaneaxial blower exhibits a linear relationship between blower output and impeller rpm. Centrifugal blowers do not have this characteristic. Therefore, with the recommended circuitry, the accuracy of a centrifugal blower system will be less than observed with the vaneaxial system.

The manufacturers of variable speed motors and controllers define the accuracy of their system in terms of speed regulation. Regulation is defined in terms of a percentage of maximum speed. Table 6-9 summarizes the characteristics of representative drives. In some cases, the manufacturer has a basic specification for speed regulation and optional circuitry for increased regulation. These options have also been identified in the table.

Table 6-9. MOTOR SPECIFICATIONS

MANUFACTURER	MOTOR TYPE	SPEED REGULATION (%)	SPEED RANGE	RATED TORQUE (NT-M) 20 HP	25 HP	30 HP
Eaton	EC	1	1695-50	33.9	33.9	44.0
	DC	2, 1	1695-88	-	-	-
Louis Allis	EC	1, 0.5, 0.1	1695-50	-	-	-
	DC	2, 1	1695-50	-	-	-
WER	EC	2, 1, 0.1	1710-50	-	-	-
	DC	2, 1, 0.1	1750-17	-	-	-
Sabina	DC	3, 1	1750-17	79.0	98.8	118.5
Randtronics	DC	2, 1, 0.5, 25	1750-0	-	-	-

In order to achieve the accuracy goal defined in the contract, at least 1 percent speed regulation must be obtained. (This assumes that the blower output is perfectly linear.) As shown in Table 6-9, this degree of regulation is available from several suppliers, using either eddy current or DC systems. The methods used to achieve this degree of control varies from manufacturer to manufacturer. Some utilize tachometer feedback control to achieve this regulation, while others use proprietary techniques to improve the system's speed regulation characteristics.

In conclusion, from the standpoint of speed regulation (i.e., accuracy), the variable-speed drive coupled to a vaneaxial blower is superior to a system employing a centrifugal blower.

6.6.2 Speed Range

The speed range required to satisfy the goals outlined in the contract is 100-to-1 or 1,750-17 rpm. This is equivalent to a wind velocity range of 100 kph to 1 kph. Table 6-9 details the speed range of the various drives.

A motor's speed range is defined at 100 percent torque, continuous duty, and torque is defined by equation (6-5)

$$\tau = I\alpha \qquad (6\text{-}5)$$

where

I is the rotational inertia and α is the angular acceleration. The required angular acceleration rate is controlled by the driving cycle and the air delivery characteristics of the fan wheel. The maximum angular acceleration is defined by equation (6-6)

$$\tau_{max} = \frac{(\text{max cycle acceleration})(\text{blower exit area})}{(\text{blower output per revolution})} \qquad (6\text{-}6)$$

Maximum torque requirements have been calculated for a representative vaneaxial unit (Aerovent 361) and centrigual blower (New York Blower 449). These are listed below.

Aerovent 361 $\tau_{max} = 27.1$ nt.-m (240 lb-in)
New York Blower 449 $\tau_{max} = 125.6$ nt.-m (1,112 lb-in)

Since maximum acceleration rates in the FTP occur at low speeds, the maximum torque demand will also occur at low motor speeds. Typical torque levels available were shown in Table 6-9. It should be noted that the eddy current motor is a torque transmitter in contrast to a DC motor which is a torque converter. As a result, the eddy current drive provides an essentially-constant torque over its entire speed range rather than torque being a function of rpm.

Two conclusions are readily apparent after reviewing the data in Table 6-9. First, eddy current drives cannot be operated at low enough rotational speeds to satisfy the 100-to-1 operating range requirement. These units could be operated at lower speeds, but overheating might occur and auxiliary cooling would require additional space and add cost and complexity. DC systems do have sufficient dynamic range to fulfill this application.

Secondly, the drives under consideration, based on power requirements, cannot provide sufficient torque to drive a centrifugal blower at the lower speed and torque imposed by the FTP. The motor can be oversized to provide sufficient torque. It is estimated that a 30 kw (40 hp) motor would be required to drive a centrifugal blower versus 15 kw (20 hp) motor to drive a vaneaxial unit.

6.6.3 Response Time

The control system which is selected for this application must be capable of tracking the motorcycle speed; both its accelerations and decelerations. As discussed in Section 6.5.2, the motors under consideration are capable of accelerating the blower. However, without external braking, the decelerations cannot be followed.

Eddy current systems are available with two different types of brakes. A friction brake can be added, but this type of unit has been designed to bring the shaft to a stop. It is not intended to slow the rotation of the unit in a controlled manner. In order to accomplish the controlled deceleration of the unit, an eddy current or electromagnetic braking system must be added. Sometimes referred to as Mutatrol, this option is available from all three known suppliers of eddy current motors, and consists of a second eddy current unit which performs the function of a power absorption unit or brake. Its braking capacity is comparable to the accelerating capacity of the eddy current motor. The cost impact of this device is examined in Section 6.6.9.

DC motors are also available with external braking packages. These auxiliary braking systems, however, are friction brakes which cannot accurately control decelerations. Controlled decelerations with a DC system can only be achieved with a regenerative DC system. These systems are effectively compact motor-generator sets.

As described by WER Industrial, a DC motor with fixed-field excitation carries an armature current which will produce torque propotional to the armature current and in the direction of the armature current. In the regenerative unit, the armature current can flow in either direction under controlled conditions, resulting in what is termed as four quadrant drive operation. In order to accomplish this, two SCR modules are placed back-to-back across the DC motor. In the motoring mode, a module will act as a rectifier to convert the incoming AC power to DC power for the motor armature. When called upon to slow the motor, the other module will function and operate as an AC line commutated inverter. By phasing on the SCRs at the proper phase angle, this module will convert the DC armature voltage to an AC wave form so that current is allowed to flow through the motor armature back into the AC line. In order to prevent the two modules from operating simultaneously, the controller should contain "lock out" or "dead band" circuitry. The regenerative DC system is ideally designed for this application, and it is the DC counterpart to Mutatrol.

As discussed in the previous section, many of these motors may have insufficient torque to accelerate a centrifugal through the expected short-term accelerations. Since the expected decelerations are of the same magnitude as the acclerations, some difficulties may be encountered in decelerating a centrifugal blower, with either a regenerative DC system or eddy current unit.

6.6.4 Delivery

Inquires were made to three manufacturers of eddy current units and seven manufacturers of DC and regenerative DC systems regarding estimates of delivery. Estimates ranged from 3 weeks to 26 weeks for all three motor styles. The manufacturers advised that delivery schedules fluctuate wildly and are usually dependent on the availability of the motor. These schedules appear to be reasonable and in line with the delivery of the other dynamometer components.

6.6.5 Size and Weight

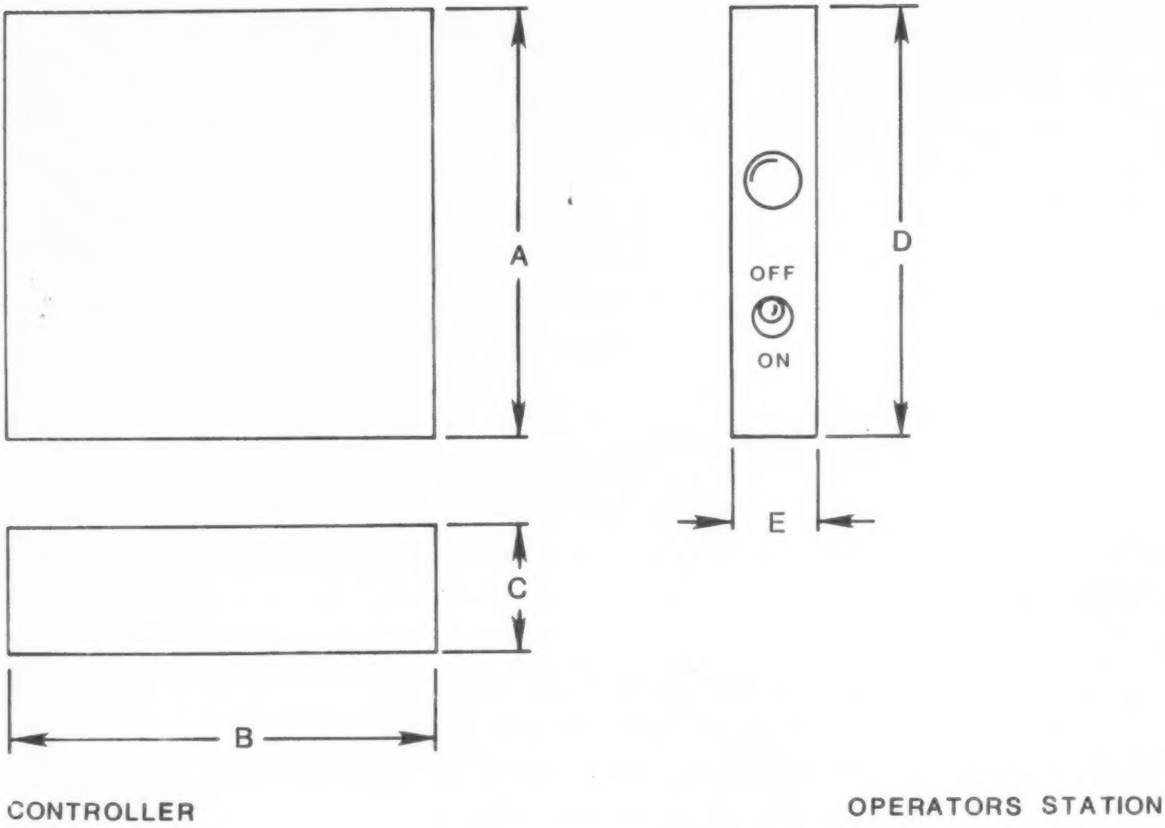
Typically, the eddy current system, DC system and the regenerative DC system consist of three components. These are the motor, the controller, and the operator's station.

Motor sizes were examined in Section 6.5.8 and comparable motor sizes were shown in Table 6-7 and 6-8. As previously discussed neither style of motor offers any significant weight or size advantage.

Figure 6-7 shows the typical dimensions of the controller and operator's station. The controller is typically wall-mounted while the operator's station is designed for hand-held operation. As shown, the eddy current controller is somewhat smaller than its DC counterpart. However, the size of the DC controller is not overly large, nor does it limit the usefulness of the DC system.

6.6.6 Adaptability

The use of an eddy current motor or DC motor in conjunction with tachometer follower circuitry and controlled braking capability, results in a versatile system capable of


A
D
OFF
ON
C
E
B
CONTROLLER
OPERATORS STATION

MANUFACTURER	MOTOR TYPE	DIMENSIONS (CM) A	B	C	D	E
Eaton	E.C.	30.5	24.1	15.3	24.1	9.1
Sabina	D.C.	91.4	76.2	30.5	24.1	7.0
Reliance	D.C.	97.8	67.3	38.7	26.1	8.6

FIGURE 6-7. CONTROLLER AND OPERATOR STATION DIMENSIONS

tracking motorcycle speeds. The limit to this tracking capability is not the controller, but rather the motor's torque/power limitations. As discussed in Section 6.5.2, less torque is required to accelerate a vaneaxial unit than a centrifugal blower. Therefore, should it be necessary to follow faster accelerations than found in the LA-4, the vaneaxial systems would be most adaptable.

In addition, most manufacturers supply tachometer follower circuitry which can be manually overidden. This would permit the blower to be operated at speeds which differ from the motorcycle speed. In this mode, studies could be conducted to examine the effects cooling has on motorcycle emissions.

Finally, the versatility of a nonregenerative DC motor is restricted because of its limited braking capacity.

6.6.7 Reliability/Maintenance

The eddy current motor requires less maintenance than a DC drive. The simple mechanical design of the eddy current drive eliminates brushes, commutators and slip-rings; all of which are used in DC drives and require periodic maintenance. In an eddy current drive, the only mechanical item which may require maintenance are the bearings between the input and output members of the magnetic clutch. The typical life of these bearings is in excess of 50,000 hours.

The superior reliability of an eddy current unit is supported by data from Louis Allis, a manufacturer of eddy current, DC, regenerative DC and variable-speed AC drives. In May 1975, their warranty expenses for eddy current drives were 0.05 percent of sales versus 0.7 percent for all types of DC systems and 4.2 percent for variable AC units.

The controllers used for both styles of drives consist of sophisticated SCR/solid-state circuitry, which is known for reliability and ease of maintenance. Periodic maintenance consists basically of periodic inspection and cleaning of the controller.

All manufacturers recommend the acquisition of a spare parts kit. These consist of spare fuses and circuit boards.

6.6.8 Interferences

The system utilized in this application should not be affected by other electrical and environmental elements in the test area. Temperature, and humidity will not be a factor since these are controlled in the test cell. Altitude does effect the performance of the blower and driver, but the altitude of Ann Arbor, Michigan, is not sufficient to pose any problems. At altitudes over 900m, modifications of blower size and motor rating must be considered, if sea level testing is to be simulated.

Electrical interference may be a problem. Kawasaki in Shakope, Minnesota, has experienced some problem with their DC system as a result of RF interferences. Consideration should be given to interference associated with high-energy capacitor discharge ignition systems. In addition, manufacturers recommend the use of isolation transformers to eliminate the effects of power-line fluctuations and warn the user to examine grounding requirements and connections.

Interference effects are difficult to foretell and are affected by the specific design of the drives. At this time, none of the drives examined appear advantageous in this respect. However, the specification should address this problem.

6.6.9 Costs

Representative prices have been obtained from several manufacturers of eddy current drives and regenerative

DC drives. Prices for DC drives have not been determined because they lack controlled deceleration capabilities. The system prices are all F.O.B. manufacturer's plant and include, as a minimum, the motor, controller with 1 percent speed regulation, tachometer follower circuitry, controlled braking capability and isolation transformer. Also, all prices are for open-type motors and do not reflect any O.E.M. discounts or price premiums for speedier delivery. Prices are shown in Table 6-10.

Table 6-10. MOTOR/CONTROLLER COSTS

MANUFACTURER	MOTOR TYPE	SYSTEM 15 KW (20 HP)	19 KW (25 HP)	COST 22 KW (30 HP)	OEM DISCOUNT
Sabina	RDC	7561	8190	8751	28%
WER Industrial	EC	-	8655	-	10%
	RDC		9109		10%
Louis Allis	EC	6147	6147		10%
	RDC	7180	8245		-

The costs of the two systems are comparable. The price spread among manufacturers appears to be greater than any spread between types of drive.

6.7 OUTLET DUCT DESIGN

The contract and NPRM define the geometrical configuration of the cooling air outlet duct. These have been summarized in Section 6.2 of this report. In this section, these specifications will be analyzed with respect to their impact on the simulation of motorcycle cooling.

The intended purpose of the cooling system is to simulate cooling conditions encountered in road operation and thereby control motorcycle engine temperatures to within ±10 percent of those observed on the road. With unlimited funds, this could easily be achieved by placing the motorcycle in a wind tunnel of sufficient magnitude. The costs to accomplish this would be substantially greater than the already high costs estimated in Sections 6.5 and 6.6. By properly configuring the outlet duct, it is believed that the simulation of road-cooling condition can be achieved within a realistic cost constraints. This will be verified during Phase II of this program.

6.7.1 Outlet Configuration of Ducting

As specified in the contract, the outlet of the cooling duct shall have a rectangular cross-section of $0.5m^2$, and the bottom of the ducting shall be located about 150mm (5.91 inches) above dynamometer level. These dimensions resulted from the physical measurement of the engine configuration of various motorcycles.

The $0.5m^2$ rectangular cross-sectional area was selected because the largest engine measured could be placed within it. If the engine were placed within a duct of this size and configuration, cooling simulation would be achieved. Unfortunately, the ducting will not encompass the engine, although the air exiting the ducting will be directed toward the engine. As shown in Figure 6-8, the velocity of the air discharged decreases at increasing distances from the duct exit. Additionally, there is substantial broadening of the discharge pattern as the distances from the exit increase. This velocity pattern will be further disrupted and broadened because the air must pass around the front wheel of the motorcycle and portions of the motorcycle frame before reaching the engine. Broadening would not be so severe during road operation because the air flow about the front wheel would be contained by the air streamlines, some distance from the side of the engine.

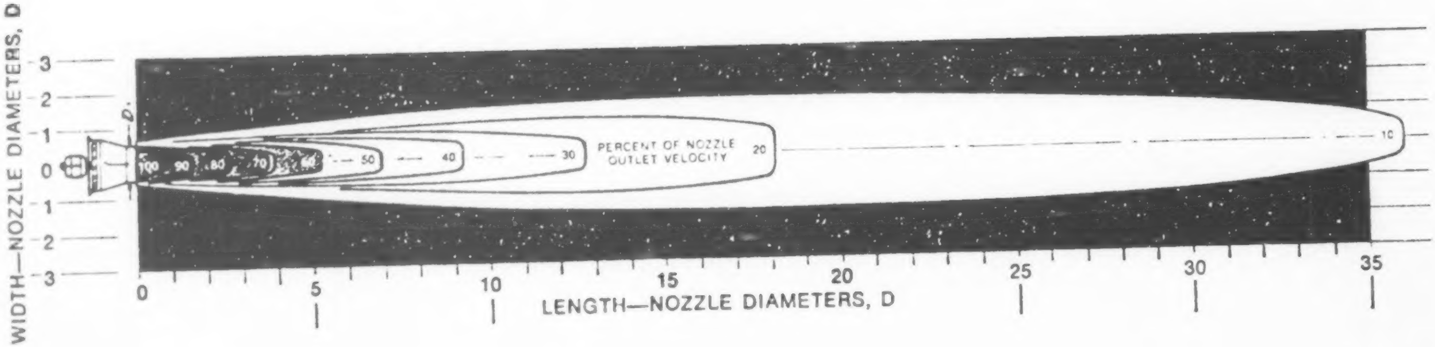


FIGURE 6-8. AIR DISCHARGE PATTERN

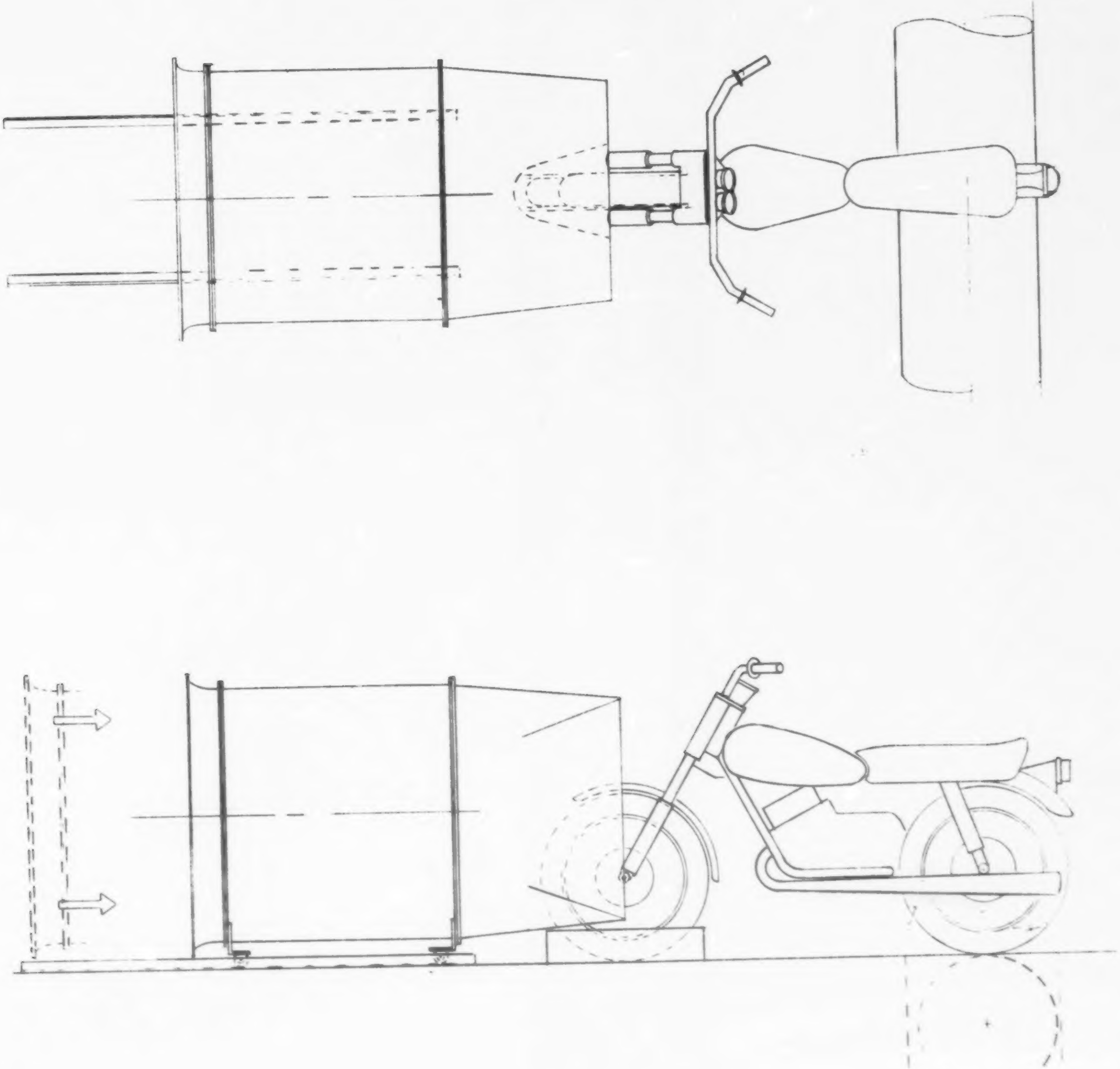


FIGURE 6-9. COOLING SYSTEM LAYOUT

The broadening effect can be minimized by designing the outlet duct to straddle or envelope part of the front wheel of the vehicle. This is shown in Figure 6-9. The effects this would have on engine-cooling are difficult to quantify because of the complex nature of the heat transfer between the engine and the air, and the various geometries of the motorcycle engine. It is expected that these effects will be analyzed experimentally during Phase II of this program.

6.7.2 Velocity Profiles

The contract defines the uniformity of the velocity profile of the air exiting the outlet duct. Figure 6-10 displays fully-developed velocity profiles for laminar and turbulent flow. As a result of the 100-to-1 range of air velocities expected, both laminar and turbulent conditions are expected. The ducting lengths will be short, however, and fully-developed flow conditions will not occur.

Figure 6-11 shows the laminar velocity profiles in a start-up condition as shown. Nearly flat velocity profiles occur as duct length decreases. The development of turbulent flow profiles is similar. Therefore, if the length of the outlet duct is less than three times the fan diameter, the velocity uniformity specification can be attained.

6.8 CONCLUSIONS

Table 6-11 summarizes the evaluation study. As quantified in the summary table, a vaneaxial system is superior to a system employing a centrifugal blower. The vaneaxial blower has distinct performance advantages resulting from its smaller physical size, characteristics, and lower cost.

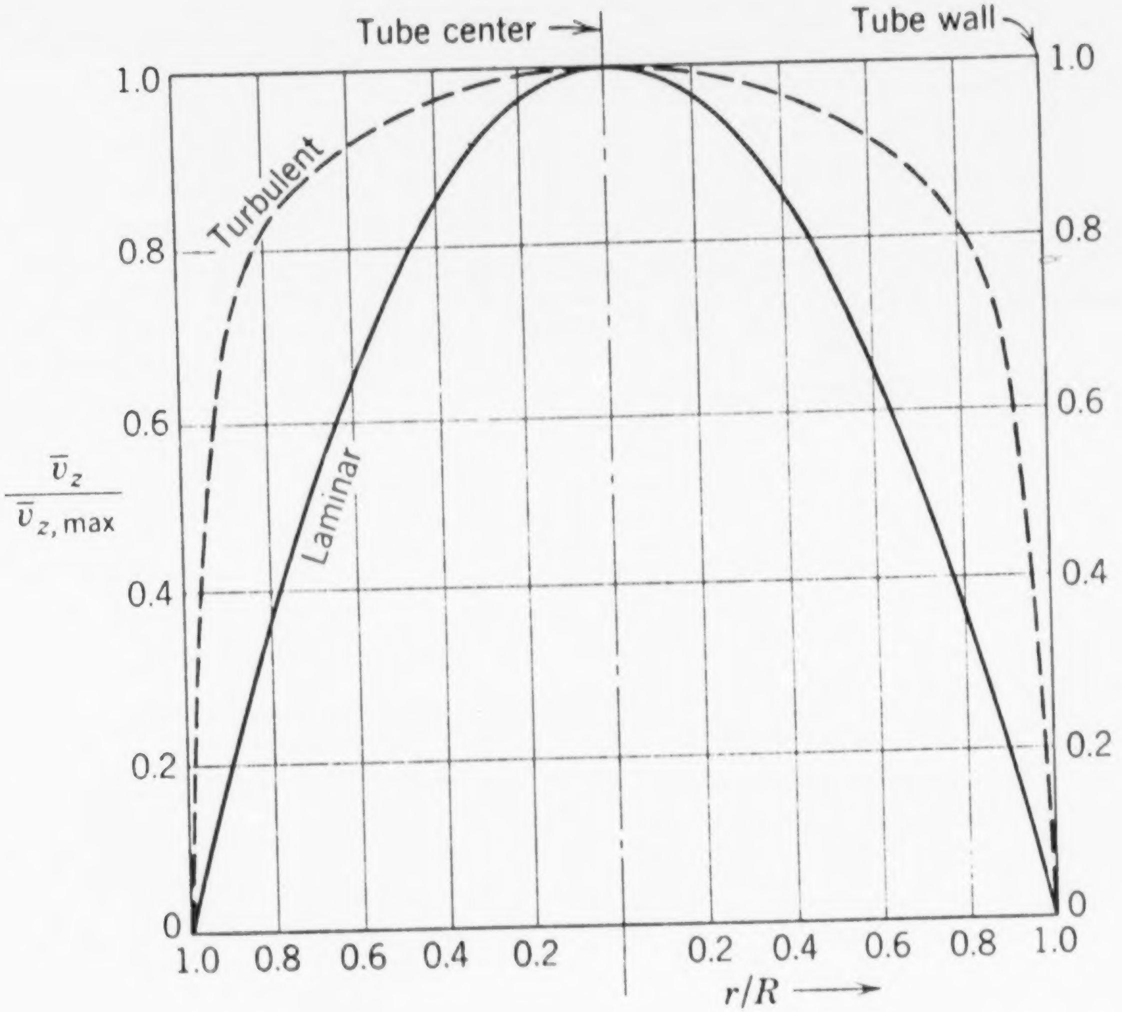


FIGURE 6-10. QUALITATIVE COMPARISON OF LAMINAR AND TURBULENT VELOCITY DISTRIBUTIONS

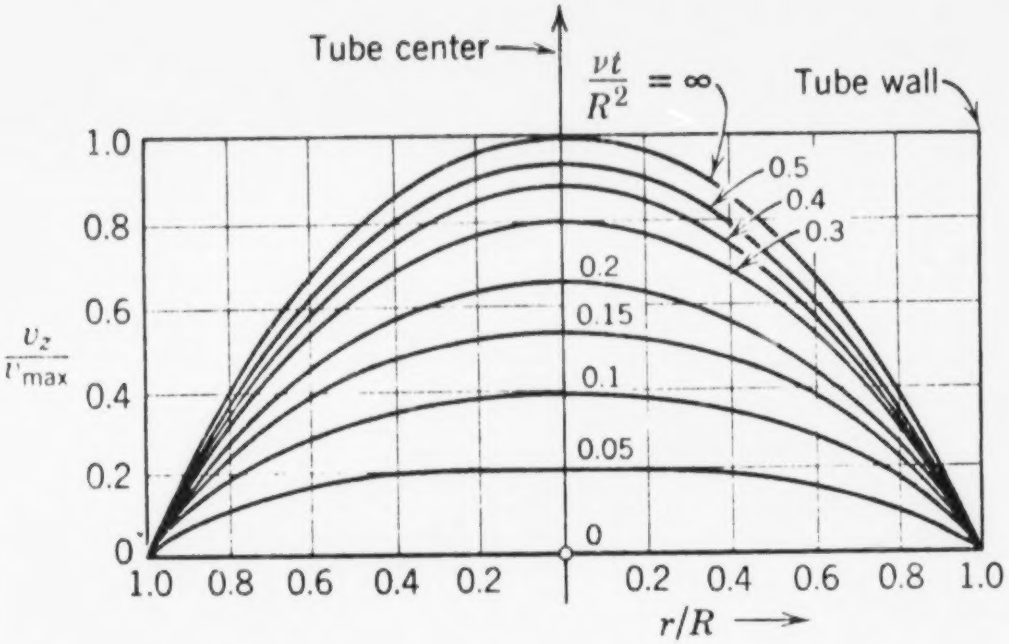


FIGURE 6-11. VELOCITY DISTRIBUTION FOR UNSTEADY STATE "START-UP" FLOW IN A CIRCULAR TUBE

Table 6-11. WEIGHTED RATINGS

CRITERION	SYSTEM NO. (SECTION 2.2)						ACCEPT. RATING 3xWT FACTOR
	Vaneaxial Blower			Centrifugal Blower			
	EC	DC	RDC	EC	DC	RDC	
I. Blower Performance							
1. Flow as a funtion of RPM and ΔP	5.0	5.0	5.0	5.0	2.0	2.0	3.0
2. Rotational Inertia	2.7	2.7	2.7	1.8	1.8	1.8	2.7
3. Delivery	2.4	2.4	2.4	2.4	2.4	2.4	2.4
4. Noise	2.4	2.4	2.4	3.2	3.2	3.2	2.4
5. Reliability/Maintenance	3.2	3.2	3.2	2.4	2.4	2.4	2.4
6. Adaptability	2.4	2.4	2.4	2.4	2.4	2.4	2.4
7. Power Consumption/Eff.	1.4	2.1	2.1	1.4	2.8	2.8	2.1
8. Motor Compatibility	2.1	1.4	1.4	2.8	2.8	2.8	2.1
9. Size and Weight	2.4	2.4	2.4	1.8	1.8	1.8	1.8
10. Flow Conditioning	2.0	2.0	2.0	1.0	1.0	1.0	1.5
11. Cost	2.0	2.0	2.0	1.5	1.5	1.5	1.5
II. Motor/System Performance							
1. Accuracy	4.0	1.0	4.0	2.0	1.0	2.0	3.0
2. Range	1.8	3.6	3.6	0.9	0.9	0.9	2.7
3. Response Time	2.7	0.9	2.7	1.8	0.9	1.8	2.7
4. Delivery	2.4	2.4	2.4	2.4	2.4	2.4	2.4
5. Size and Weight	2.1	2.1	2.1	2.1	2.1	2.1	2.1
6. Adaptability	2.8	0.7	2.8	1.4	0.7	1.4	2.1
7. Reliability/Maintenance	2.8	2.1	2.1	2.8	2.1	2.1	2.1
8. Interferences	2.1	2.1	2.1	2.1	2.1	2.1	2.1
9. Costs	2.1	2.1	2.1	2.1	2.1	2.1	2.1
Total Rating	50.8	45.0	51.9	40.3	38.4	41.0	45.6
Ranking	2	3	1	5	6	4	

The evaluation also indicates the regenerative DC drives and eddy current motor/brake systems are suitable for this application. The first of these options can meet the 100-to-1 dynamic range requirement but it is felt that a 30-to-1 range of performance should be acceptable for this application. Assuming this smaller range is acceptable, then the eddy current system offers slight size, maintenance and cost advantages. A nonregenerative DC drive is unacceptable because it would be unable to follow motorcycle decelerations.

Section 7

TASK 7 - DYNAMOMETER SYSTEM ANALYSIS

7.1 INTRODUCTION

In this report a cost-effectiveness study comparing three alternative dynamometer designs will be presented. The performance characteristics of these alternatives will be compared to the performance requirements developed in previous task reports and detailed in the Notice of Proposed Rulemaking (NPRM) and the contract. As a result of this study and evaluation, a design will be recommended. A specification for this recommended system will, subsequently, be developed.

7.2 REVIEW OF REQUIREMENTS

Paragraph 85.478-15 of NRPM defines the inertia and road-load requirements for a motorcycle dynamometer. These requirements are more explicitly defined in the contract and the Task 2 Report.

As stated in the contract, the dynamometer should be capable of simulating inertia in the range from 100 kg (220 lbs) to 700 kg (1,543 lbs). The simulation should provide discrete inertia steps with intervals of 10 kg (22 lbs).

Road-load data on ten motorcycles was provided by the Project Officer to Olson Laboratories for analysis. The analysis was presented in Task 2. It indicated that at motorcycle speeds of 110 kph, which was the maximum speed encountered during the durability cycle, a maximum power absorption capacity of 14.7 kw (20 HPm) was required. This assumes that flywheels will be used for inertia simulation. If inertia is to be simulated electrically, the absorption capacity would increase to approximately 37 kw (50 HPm), and an additional 18.4 kw (25 HPm) motoring capacity would be needed.

As discussed in the Task 2 Report, the road-load characteristics of various motorcycles differ substantially, and one characteristic curve is insufficient to simulate road-load conditions of all motorcycles. This can be achieved by using a controller which can accept the algorithm developed by the EPA and which loads the power absorption unit (PAU), as required. These requirements must be satisfied by the alternative selected.

7.3 REVIEW OF RECOMMENDATIONS FROM TASKS 2, 3, AND 4

In Tasks 2, 3, and 4, evaluations of PAU roller configurations and inertia simulation techniques were presented, and recommendations were made. These findings were discussed in detail at a review meeting held with the Project Officer and other members of the EPA staff on May 21, 1975. Subsequently, in a letter dated June 18, 1975, the Project Officer directed Olson to examine three systems. The three systems are:

1. Eddy current PAU - inertia simulation by mechanical (flywheel) means.
2. DC or AC motor generator - inertia simulation by mechanical (flywheel) means.

3. DC or AC motor generator - partial mechanical plus electrical simulation using the motor generator.

Additionally, each of the three systems would employ a smooth, single dynamometer roll having a diameter of 530.5 mm

Olson Laboratories concurs that these alternatives are viable approaches which would result in systems which achieve the design goals outlined in Section 7.2.

7.4 DESIGN ALTERNATIVES

As stated in Section 7.3, the Project Officer has requested that the three design alternatives be examined in detail. As a result of this study, a final design will be developed and specifications will be formulated. In this section, evaluation criteria will be defined. Subsequently, a cost-effectiveness study of the design alternatives will be conducted.

In reality, there are five systems which should be considered. These combine the three types of power absorbers with the two methods of inertia simulation. The five combinations are:

1. DC motor generator - inertia simulation by mechanical (flywheel) means.

2. DC motor generator - inertia simulation by partial mechanical plus electrical techniques.

3. Eddy current PAU - inertia simulation by mechanical (flywheel) means.

4. AC motor generator - inertia simulation by mechanical (flywheel) means.

5. AC motor generator - inertia simulation by partial mechanical, plus electrical techniques.

In the Task 2 analysis, the characteristics of the AC motor generators were examined. As reported, adjustable-speed AC drives are advantageous in applications; wherein sparkless operation, ultra high speed, multiple-drive synchronization, and/or adverse environmental conditions are a factor. None of these factors seem to be considerations in this application. Additionally, Louis Allis, a manufacturer of both AC and DC systems, reports that costs of an AC system are twice that of a comparable DC system. Therefore, further considerations of AC systems seem unwarranted. Accordingly, this evaluation will be limited to those alternatives which use eddy current or DC PAU's.

7.4.1 Evaluation Criteria and Ranking System

In order to arrive at the most cost-effective design for a motorcycle dynamometer, the various combinations of system components must be systematically and comprehensively evaluated and compared. The analytical procedures must consider all factors and characteristics that contribute to system performance and costs, influence system operational procedures, and affect system maintenance, maintainability, and reliability.

All of the alternatives will be evaluated with respect to each of the criteria and will be rated on a scale of 1-to-5; where 1 is unacceptable, 3 is acceptable, and 5 is outstanding. Additionally, a weighting factor has been designated for each of the criteria, defining its relative importance. Each of the ratings will be adjusted by using the weighting factor, and the rankings of the alternatives will be calculated by summing the weighted ratings.

The evaluation criteria, their respective weighting factors and the resultant ratings are listed in Table 7-1.

7.5 EVALUATION OF DYNAMOMETER SYSTEMS

7.5.1 DC PAU - Inertia Simulation by Partial Mechanical Plus Electrical Techniques

Performance

In this configuration the system would consist of a regenerative-type DC motor which functions as the PAU, a controller, and a single flywheel having an inertia of 300 kg, midway between the 100 kg and 700 kg limits, specified in the contract. Prior to the test, the operator would input into the controller the mass of the motorcycle, and the constants of an algorithm which define the road-load characteristics of the vehicle being tested.

During the test, the controller would provide signals to the DC motor based upon the input signals to the controller and the characteristic data inputed to the controller prior to the test. The continuous inputs which are fed to the controller are load, as measured by a load-sensing element or torque bridge, and vehicle velocity, as measured by a tach generator or optical encoder mounted on the shaft of the dynamometer roll. The controller would calculate the road-load from the algorithm and compare it to the load as measured by the load cell. The resulting error signal would be used to provide a signal to the DC motor, which would result in additional power absorption or motoring. It is estimated that 100 milliseconds would be required to perform this comparison and generate the error signal. This

Table 7-1. SYSTEM ANALYSIS FACTORS

FACTORS	WT. FACTOR	DYNAMOMETER CONFIGURATION RATING					
		DC System Elect. Inertia		DC System Mech. Inertia		Eddy Current Mech. Inertia	
		Unweighted	Weighted	Unweighted	Weighted	Unweighted	Weighted
I. Performance							
1. Ability to Simulate Road Conditions	1.0	2	2.0	4	4.0	4	4.0
2. Ability to Simulate Vehicle Inertia	1.0	2	2.0	4	4.0	4	4.0
3. Ability to Accommodate Various Morotcycles	1.0	3	3.0	3	3.0	3	3.0
4. Fail-Safe Provisions	1.0	3	3.0	3	3.0	3	3.0
5. Ability to Simulate Durability Cycle	0.8	3	2.4	3	2.4	4	3.2
6. System Repeatability	1.0	2	2.0	3	3.0	3	3.0
II. Operation							
1. Operating Personnel Training	0.5	3	1.5	3	1.5	3	1.5
2. Pre-Test, Set-Up Procedures	0.6	2	1.2	3	1.8	3	1.8
3. Supporting Utilities Required	0.6	4	2.4	3	1.8	2	1.2
4. Test Data Acquisition	0.4	3	1.2	3	1.2	3	1.2
5. Facility Spatial Requirements	0.5	4	2.0	2	1.0	2	1.0
III. Maintenance							
1. System Complexity	0.6	3	1.8	3	1.8	4	2.4
2. Maintainability of System	0.8	3	2.4	3	2.4	3	2.4
3. Reliability of Components	0.8	3	2.4	3	2.4	3	2.4
4. Preventative Maintenance	0.6	3	1.8	3	1.8	3	1.8
IV. Costs							
1. Purchase Price	0.8	2	1.6	1	0.8	3	2.4
2. Installation Costs	0.5	3	1.5	3	1.5	3	1.5
3. Annual Operating Costs	0.6	3	2.4	3	2.4	3	2.4
Cumulative Ratings			38.4		41.6		45.0
Ranking			3		2		1

time is based on a 10-millisecond computation time and a 100-millisecond period for sensing and averaging the load signal. The picking and averaging time would occur concurrently with the calculation time. The system response time, defined as the elapsed time between signal input to the controller and actual armature current change resulting in PAU load change, would be on the order of 500 to 1,000 milliseconds. This response time is controlled by the inertia of the motor armature and the controller design, which would vary with motor design.

By utilizing a flywheel having an equivalent inertia approximately at the mid-point of the inertia range, the motor will only be required to simulate half of the total inertia spread. For vehicles having inertias greater than the mid-point value, the DC regenerative motor would operate in a regeneration (absorption) mode; thus, adding to the simulated inertia seen by the motorcycle. For those motorcycles with an inertia mass less than that of the fixed flywheel, the DC motor would function in both motoring and absorption modes. As a result of having to function in both modes, response times may be increased, and road-load simulations for the smaller vehicles may not be as accurate as the simulation for the larger vehicles. In order to have a transition from a motoring mode to an absorption mode, the system response time cited above may be increased by an additional 500 milliseconds. This added time will be affected by the inertia of the motor's armature and the characteristics of the controller.

The addition of more than one flywheel, decreases the capacity for total inertia simulation upon the motor. Thus, the motor will work more in a regenerative mode than in a motoring mode. This will effectively reduce the size requirement of the original motor selection, but, concurrently, it will require the addition of flywheels, bearings, subframe, clutching and de-clutching mechanisms. The cost for these mechanisms must be balanced against the differential cost of a smaller DC regenerative motor.

In this application, repeatability of operation is critical. The degree of repeatability will be affected by the stability and repeatability of the road-load and velocity measurements and the controller. The use of the load cells are recommended because they are accurate, rugged and unaffected by temperature fluctuations. Armature current measurements can be used to determine motor load, but the accuracy of these measurements are affected by motor temperature. It has been estimated by Sabina Electric, a manufacturer of regenerative DC systems, that repeatability within 0.1 percent can be achieved utilizing an optical encoder for measuring dynamometer roll speeds and a temperature-compensated load cell for power measurements.

Fail-safe provisions are standard in most DC systems. Accordingly, the DC regenerative motor system is typically equipped with an auxiliary cooling fan that is connected to the motor and force-ventilates the motor, but does not add significant inertia to the system. Additionally, the controller incorporates an overload system which allows for low speed, high torque operation without the danger of overload, and is equipped with an armature current fuse protection. This latter has an adjustable set point and limits the current to a specific level. Typically, it can be set between 125 and 200 percent maximum current rating. The controller also has an electronic overload switch which trips out the motor when it sees an instantaneous overload in excess of the set point. Additional protection is provided against excessive acceleration or deceleration.

In summary, with respect to performance, the DC system, utilizing electrical techniques to simulate inertia, can approximately simulate road conditions. The simulation will be limited by the response-time characteristics of the system which is affected by motor design.

By properly sizing the regenerative system, the dynamometer can be used for both durability and certification testing, and its use will not be affected by the size or performance characteristics of commercially-available "street" motorcycles.

Operation

The controller for this system will be quite complex, but the unit itself can be operated by a trained technician. The level of training will probably be more extensive than that required to operate a Clayton hydrokinetic unit and will probably be less than the training required to operate a programmable, automatic light-duty vehicle durability dynamometer such as produced by Labeco.

Pre-test set-up procedures will not be extensive. They will basically consist of placing the vehicle on the dynamometer, restraining it, and dialing into the controller the appropriate inertia setting and road-load algorithm constants. Since the unit requires no utilities except power, typically 460 VAC, 60 Hz, 3 Ø, no other connections have to be made.

One pre-test procedure which must be periodically performed is the calibration of the dynamometer. Calibration procedures for dynamometers equipped with flywheels are straightforward and well defined. However, procedures for verifying the inertia simulation accuracy of this system have not been fully defined. These procedures, when developed, may be quite complex, requiring additional manpower and equipment.

As in all the sytems being examined, a driver's aid must be provided. At a minimum, it will provide data on instantaneous speed and horsepower. Additional readouts such as armature current may be useful, but none are mandatory.

Of the three configurations being examined, the DC system with electrical inertia simulation will occupy the least amount of space. Although the DC motor is larger than an eddy current brake, the lack of a flywheel system results in a net decrease in space requirements. Of course, the pit depth and width will be a function of roller length and DC motor frame length. The latter will vary from manufacturer to manufacturer.

Maintenance

The system proposed here is complex. A DC regenerative system with SCR-type controller, will consist of a velocity signal generator mounted on the roll, a load cell, and an operational box which will use the voltage signal from the load cell to provide the proper force signal to load the PAU. This operational box can be in the form of a mini-computer or microprocessor.

The components of this DC system are all designed as standard components but not as a total system. There are no electronic components that are specifically designed for this application. All operational amplifiers are standard industry type. All components are shelf items with the exception of enclosures, etc.

Preventive maintenance is required to maintain the excellent performance characteristics of a DC motor. Items requiring periodic maintenance include brushes, commutator and stator. Brushes should be replaced every 5,000 hours of operation. Of course, undersized or oversized brushes will have a detrimental effect on the life of the brush. The brush selection is based upon the application and must be determined in the intitial design and tested in the field after the first or second brush change-out.

With the exception of the replaceable electrical motor components, a realistic life of the system is 10 years with recommended maintenance on the motor. Recommended maintenance includes motor teardown every 3 years. This includes cleanout, bearing replacement and commutator dress up. Periodic brush replacement should occur approximately every 3 months. With the exception of human error or mechanically caused problems in the electronics, reliability of the electronic components should be high. Since most of the electronics are self-compensated for drift component, age is not a problem.

Costs

Tables 7-2 and 7-3 summarizes the prices of various DC and eddy current dynamometers. These prices were provided by Meidensha, Ltd., and provide valuable information regarding the relative costs of the alternative configurations. A comparable estimate was received from Consine Dynamics, Ltd. As shown in Table 7-2, the DC system with electrical inertia simulation is the least expensive of all the configurations being examined. However, as shown in Table 7-3, the cost impact of enlarging the power absorption capacity to fulfill the requirements of the durability cycle is significant, because the capacity of the regenerative system must be doubled. The installation costs of the three systems are comparable.

Typically, the components described above are available within 3 to 6 months after an order is placed.

7.5.2 DC System with Mechanical Inertia Simulation

Performance

This system is configured similarly to the DC system utilizing electrical inertia simulation. The basic difference is the addition of six flywheels having equivalent inertias of 10, 20, 40, 80, 160, and 320 kg.

Table 7-2. PRICE AND SPECIFICATION FOR EMISSION DYNAMOMETER

	ROLLER	FLYWHEEL	DYNAMOMETER WITH CONTROLLER
(A1) D.C. Chassis Dynamometer with Flywheels	Steel made, 530.5mm	10kg+20kg+40kg +80kg+150kg direct control	D.C. Cradled Dynamometer with Controller, 15hp absorption, 11hp motoring 1,000rpm-100km/h)
Total Price: $48,150.00	$3,650.00	$20,300.00	$24,200.00
(A2) D.C. Chassis Dynamometer with two Flywheel and 70kg Electrical Intertia Simulation	"	80kg+150kg direct control	"
Total Price: $38,950.00	$3,650.00	$10,000.00	$25,300.00
(A3) D.C. Chassis Dynamometer with one Flywheel and 150kg Electrical Inertia Simulation	"	150kg remote control	D.C. Cradled Dynamometer with Controller, 17hp absorption, 13hp motoring 1,000rpm (100km/h)
Total Price: $33,950.00	$3,650.00	$4,600.00	$27,700.00
(A4) D.C. Chassis Dynamometer with 300kg Electrical Inertia Simulation	"	Fixed Flywheel 180kg including roller and dynamometer inertia mounted on roller shaft	D.C. Cradled Dynamometer with Controller 20hp absorption, 15hp motoring 1,000rpm (100km/h)
Total Price: $31,270.00	$3,650.00	$770.00	$26,850.00
(A5) Eddy Current Chassis Dynamometer with Flywheel only	"	10kg+20kg+40kg +80kg-150kg direct control	15hp absorption 1,000rpm model TW-55 cooling water 6L/min
Total Price: $38,950.00	$3,650.00	$20,300.00	$15,000.00

Table 7-3. PRICE AND SPECIFICATION FOR DURABILITY DYNAMOMETER

	ROLLER	FLYWHEEL	DYNAMOMETER WITH CONTROLLER
(B1) D.C. Chassis Dynamometer with Flywheels	Steel made, 530.5mm	10kg+20kg+40kg +80kg+150kg direct control	D.C. Cradled Dynamometer with Controller, 25hp absorption, 19 motoring 1,200rpm (120km/h)
Total Price: $50,050.00	$3,650.00	$20,300.00	$26,100.00
(B2) D.C. Chassis Dynamometer with two Flywheel and 70kg Electrical Intertia Simulation	"	80kg+150kg direct control	"
Total Price: $40,500.00	$3,650.00	$10,000.00	$26,850.00
(B3) D.C. Chassis Dynamometer with one Flywheel and 150kg Electrical Inertia Simulation	"	150kg remote Ccontrol	D.C. Cradled Dynamometer with Controller, 45hp absorption, 33hp motoring 1,200rpm (120kg/h)
Total Price: $40,150.00	$3,650.00	$4,600.00	$31,900.00
(B4) D.C. Chassis Dynamometer with 300kg Electrical Inertia Simulation	"	180kg including roller and dynamometer inertia	D.C. Cradled Dynamometer with Controller, 50hp absorption, 37hp motoring 750/1,200rpm (75/120km/h)
Total Price: $42,830.00	$3,650.00	$770.00	$38,400.00
(B5) Eddy Current Dynamometer with Flywheel only	"	10kg+20kg+40kg +80kg+160kg direct control	Eddy Current Cradled Dynamometer with Controller, 25hp absorption 1,200rpm Model TW-55 cooling water 9L/min.
Total Price: $38,950.00	$3,650.00	$20,300.00	$15,000.00

There are several advantages to utilizing flywheels which improve the performance of this dynamometer, as compared to a dynamometer using electrical inertia simulation. These include:

- Operation of the DC system only in the regenerative mode resulting in faster response times.
- Exact inertia simulation, regardless of vehicle speed or mode of operation.
- Smaller power absorption capacity.
- Improved performance repeatability.
- Motoring capability.

When using flywheels to simulate inertia, it may be necessary at times to motor the dynamometer to compensate for windage and friction losses in the flywheel system. Clayton has estimated that load loses as great as 3 kw may occur, depending on the number of flywheels engaged. This loss may be comparable to the motorcycle loads which occur at low speeds. The compensation method would require the determination of the system-loss characteristics as a function of rpm for each of the 60 flywheel combinations. A driving torque could then be applied by the motor according to a predetermined program based upon instantaneous rpm and the position of the inertial selection switch. This system would not be complex and would not require any modification to the motor and only minimal controller modifications.

As a result of the addition of flywheels, additional fail-safe provisions would be added. These include enclosing the flywheel assembly and installing an interlock to prevent engagement or disengagement of flywheels during dynamometer operation or in the event of power or air interruption. There would also be an overspeed warning system.

Operation

The operation of the two DC systems is almost identical. Personnel training efforts would be comparable, and data acquisition requirements and set-up procedures would be identical. Calibration procedures would be simplified.

A shop air supply would be required to operate the flywheel engagement system. This requires a compressed air supply having a minimum pressure of 413.7 kPA (60 psig).

Power requirements are comparable to the other DC system, but AC current requirements might be slightly reduced. Provisions must be made to return the regenerated power to the line.

Calibration procedures, such as coastdown-tests, are well-defined for dynamometers equipped with flywheels and can easily be adapted for use with this sytem. Since the dynamometer possesses motoring capabilities, the coastdown procedure can be easily automated for little additional cost.

Spatial requirements are maximized with this system. This occurs because the flywheels are significantly larger than the roller and DC motor.

Maintenance

The complexity of the controller is reduced by utilizing flywheels for inertia simulation, and this should result in a more reliable system. However, with the introduction of the flywheels, additional maintenance requirements are necessary. These include periodic lubrication and the inspection and possible replacement of bearings. The level of maintenance is comparable to the other DC configuration.

Costs

The purchase price of a DC system with flywheels is significantly higher than the other DC system. This occurs even though the capacity of the DC systems using flywheels is significantly lower as shown in Tables 7-2 and 7-3. The added costs of the flywheels more than offset the reduced costs of the smaller regenerative motor. As with the other DC system, there is a substantial cost differential between a certification dynamometer and a durability dynamometer.

Installation and maintenance costs of the two DC systems are comparable. However, the operational costs of the DC-flywheel system will be slightly lower than the costs of the DC-electrical inertia system because of the lower power requirements of the former systems.

Components of this system are standard products, but are not necessarily stock items. Estimated component delivery times are 12 to 26 weeks after orders are placed.

7.5.3 Eddy Current PAU with Flywheels

Performance

The eddy current system consists of an eddy current brake, flywheel assembly, and controller. The controller and flywheel assembly function in a similar manner to the components used in the DC-flywheel systems. The controller will provide a DC-excitation signal for adjusting the clutch field coil on the eddy current brake.

The response times of the eddy current unit will be slightly slower than its DC counterpart. The computation and picking times within the controller will be on the order of 100 milliseconds, and the response time of the PAU to changes in the excitation signal is estimated to the 500 to 1,000 milliseconds. However, quantitative information on the response-time characteristics of eddy current brakes could not be obtained from manufacturers.

The power absorption characteristics of an eddy currrent absorber at low rotational speeds, less than 100 rpm, is critical in this application. Available data is skimpy. The Lear Siegler eddy current brakes, examined in Section 2, are not effective power absorbers at these conditions. On the other hand, data published by the Dynamatic Division of Eaton Corporation in their publication, "Dynamometer Handbook of Basic Theory and Applications," indicates that eddy current brakes can be effective absorbers at low rotational speeds. A typical set of torque curves is shown in Figure 7-1.

The eddy current flywheel system is equipped with comparable fail-safe provisions as the other configurations. Should a water-cooled brake be used, as is recommended below, additional interlocks willl be required to protect against water/temperature failures.

The repeatability of the system's performance is imperative. The use of flywheels assures the exact simulation of inertia. In order to ensure the repeatable performance of the eddy current unit, it should be water-cooled. This eliminates performance fluctuations caused by motor and rotor temperature changes. The repeatability of the controller will be less than 0.5 percent.

Operation

The major difference between the operation of this system and that of the other two systems is the requirement for water cooling. A typical cooling configuration is shown

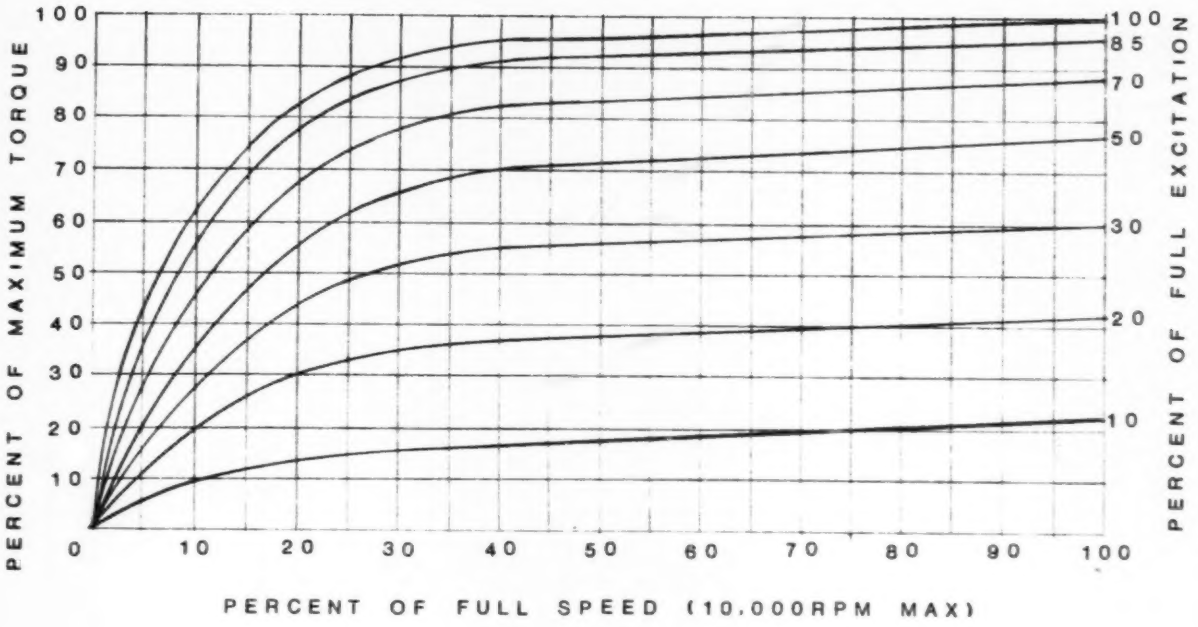


FIGURE 7-1.

TYPICAL TORQUE CURVE OF STANDARD DYNAMATIC DYNAMOMETER (EDDY-CURRENT TYPE) in Figure 7-2. A closed-loop, water-to-water heat exchanger is recommended in order to minimize thermal loads in the test cell.

Personnel training and data acquisition requirements would be similar to those required in the other configurations. Additional steps would have to be added to the set-up procedures to ensure that the cooling water system had been started. The eddy current brake is only a power absorber; it is incapable of motoring. Should a motoring mode be required so as to perform automated coast-down tests or windage compensation, an eddy brake can be coupled to an eddy current motor. This system is available from both Eaton and Louis Allis. The cost/performance of the eddy current motor/absorber versus a regenerative DC drive have been examined in Section 6.

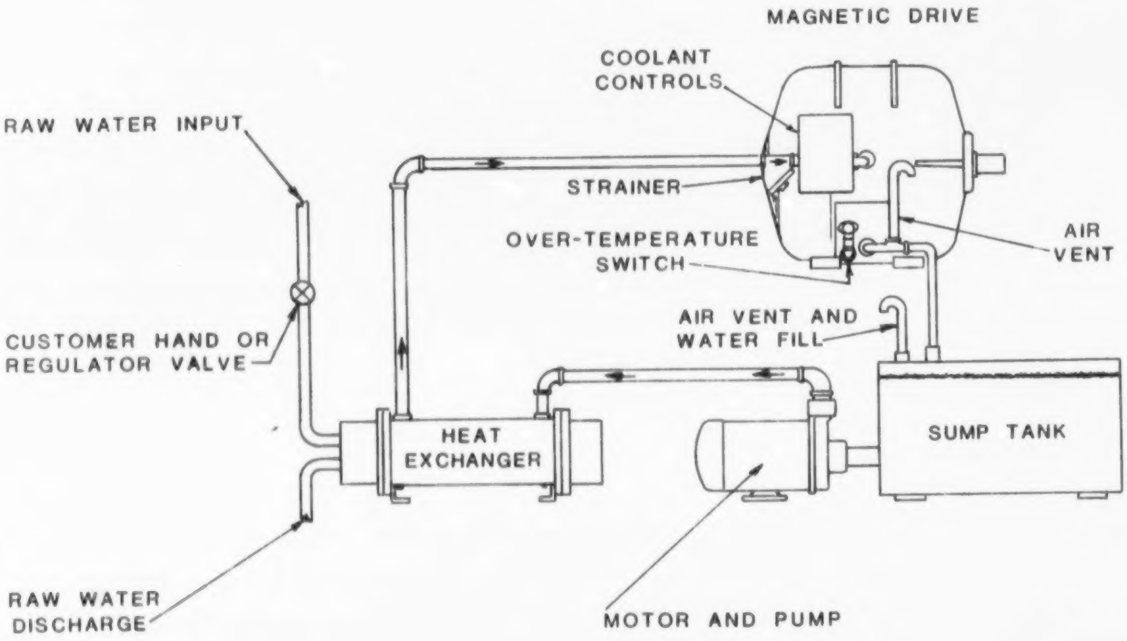
Spatial requirements would be comparable to the DC-flywheel system. While the eddy current brake is smaller than the DC unit, added space will be required for the heat exchanger and related plumbing.

Maintenance

The eddy current system is the least complicated of the three alternatives. The brake unit has less moving parts than a comparable DC unit. There are no brushes, commutaters, etc., which require replacement. There are bearings in the unit which require periodic lubrication.

The major maintenance item in this design is the cooling system. The cooling system is required to dissipate heat from the eddy current brake. A closed-loop system with a water-to-water heat exchanger is recommended in order to minimize water usage, which is approximately 644 liters/min/kw (12.7 gal/min/hp), and reduce the heat loss to the laboratory environment. Since the water and its natural contaminants can corrode the internal parts of the eddy current unit, certain precautions should be taken. These include:

WATER TO WATER HEAT EXCHANGERS


MAGNETIC DRIVE
COOLANT CONTROLS
RAW WATER INPUT
STRAINER
AIR VENT
OVER-TEMPERATURE SWITCH
AIR VENT AND WATER FILL
CUSTOMER HAND OR REGULATOR VALVE
HEAT EXCHANGER
SUMP TANK
RAW WATER DISCHARGE
MOTOR AND PUMP

PIPING SYSTEM FOR A MAGNETIC DRIVE WITH HEAT EXCHANGER

FIGURE 7-2. EDDY CURRENT BRAKE COOLING SYSTEM

- Coating the inner surfaces of the brake with an epoxy coating.

- Installation of strainers to remove particulate matter.

- Use of 50 percent water and antifreeze solution to minimize corrosion.

With the proper maintenance, the eddy current power absorber has a useful life in excess of 20 years.

Costs

As shown in Tables 7-2 and 7-3, the eddy current system is the least expensive of the three alternatives. There is no cost penalty incurred if the system is designed to meet the objectives of the more demanding durability cycle. A severe cost penalty is introduced if a motoring capability is required. Then, the costs are compatible to a DC system with flywheels.

Installation and operating costs are comparable to the other alternatives examined. Also, delivery times are consistent with the other alternatives.

7.6 CONCLUSIONS AND RECOMMENDATIONS

The results of this study are tabulated in Table 7-1. As shown there, the eddy current flywheel system is ranked first. The second-ranked alternative is a DC system employing flywheels for inertia simulation. The least attractive method is a DC system utilizing both electrical and mechanical techniques for inertia simulation.

With respect to performance, the DC system with flywheels and the eddy current system with flywheels are comparable. The latter system has been recommended because of its lower price, reduced maintenance requirements, and higher reliability. Should motoring capability be required, these advantages would be virtually eliminated. There is no technical information in this study which justifies the exclusion of a DC-flywheel system.

The use of electrical techniques for simulating inertia would be acceptable for use in a mileage accumulation dynamometer. However, it should not be used in the certification dynamometer because of its lengthened response time characteristics, potential inaccuracies which will vary with the motorcycle mass, and complex calibration procedures. Since flywheels are a primary inertia standard, there is no justification for simulating inertia by electrical means.

The discussions which have been presented in this report have been qualitative in nature. Throughout this program, dynamometer manufactureres have been contacted in an effort to obtain quantitative information on which our recommendations could be based. The data have not always been forthcoming. It is probable that, for the most part, the data have never been generated. If the information is known, it is considered proprietary by the manufacturers. However, for the purpose of recommending a design approach, it is believed that sufficient information was obtained to develop a firm qualitative analysis.

Appendix A

MOTORCYCLE TEST SUMMARY

MOTORCYCLE TEST SUMMARY

NOTE: 1) DATA CORRECTED FOR 1/2 OF ZERO SHIFT.
2) LIKE LETTERS DENOTE APPROX. SAME MC LOAD INDEX.

CONSTANTS: HONDA WHEEL RPM = (COUNT X 1.3333)
HARLEY D. WHEEL RPM = (COUNT X 1.1765)

FAU ROLL RPM, 8.65D = (MPH X 38.86)
" " " 12.75D = (MPH X 26.36)
" " " 20.00D = (MPH X 16.81)

26MAR75 PHW
NC # 323
SHEET 1 OF 5

TEST IDENTIFICATION

TEST #	DATE	AMB. TEMP (°F)	MOTORCYCLE	NO. OF ROLLS	ROLL DIA. (IN)	ROLL SPACE ℄ TO ℄ (IN)	TIRE PRESS (PSIG)	PAU SPEED ROLL (RPM)	PAU TORQUE (#FT)	PAU POWER (HP)	MC SPEED (RPM)	MC TORQUE (#FT)	MC POWER (HP)	Δ TORQUE MC - PAU (#FT)	Δ POWER MC - PAU (HP)	LOAD INDEX
I	12MAR75	—	HONDA 90	2	8.65	15.56	14	416	0.85	0.07	139	16.00	0.42	15.15	0.35	A
								1026	1.85	0.36	415	19.40	1.53	17.55	1.17	A
								1780	3.65	1.24	716	24.80	3.38	21.15	2.14	B
								1127	15.15	3.25	465	51.60	4.57	36.45	1.32	C
								719	—	—	289	14.00	0.77	—	—	-
								1088	—	—	437	14.40	1.20	—	—	-
								1146	—	—	459	13.60	1.19	—	—	-
								1484	—	—	592	15.00	1.69	—	—	-
								1858	—	—	740	15.20	2.14	—	—	-
								459	—	—	183	13.80	0.48	—	—	-
II	13MAR75	-	HONDA 90	2	8.65	15.56	14	459	—	—	183	14.40	0.50	—	—	-
								1150	—	—	463	15.40	1.36	—	—	-
								1570	—	—	628	13.12	1.57	—	—	-
								1640	3.35	1.05	660	22.12	2.78	18.77	1.73	B
								1177	14.45	3.24	487	52.20	4.84	37.75	1.60	C
1	14MAR75	—	HONDA 90	2	8.65	15.56	14	1605	5.85	1.79	648	31.12	3.84	25.27	2.05	B
								1162	14.95	3.31	481	54.08	4.95	39.13	1.64	C
								385	—	—	156	12.80	0.38	—	—	-
								1041	—	—	419	13.72	1.09	—	—	-
2	14MAR75	—	HONDA 90	2	8.65	15.56	14	392	—	—	160	12.80	0.39	—	—	-
								1073	—	—	431	14.08	1.16	—	—	-
								1613	6.35	1.95	653	30.20	3.75	23.85	1.80	B
								1170	15.00	3.34	483	50.48	4.64	35.48	1.30	C
3	14MAR75	—	HONDA 90	1	8.65	—	20	1554	8.35	2.47	635	29.20	3.53	20.85	1.06	B
								1189	16.30	3.69	489	45.88	4.27	29.58	0.58	C
								381	—	—	156	9.60	0.29	—	—	-
								1006	—	—	408	11.12	0.86	—	—	-
4	14MAR75	70	HONDA 90	2	8.65	15.56	20	381	—	—	151	11.60	0.33	—	—	-
								940	—	—	371	12.80	0.90	—	—	-
								1655	6.35	2.00	660	30.00	3.77	23.65	1.77	B
								1161	15.40	3.46	481	51.68	4.73	36.28	1.27	C

27 MAR 75 PHW
W.O. # 333
SHEET 2 OF 5

MOTORCYCLE TEST SUMMARY

CONSTANTS: HONDA WHEEL RPM = (COUNT X 1.3333)
HARLEY D. WHEEL RPM = (COUNT X 1.1765)

FAU ROLL RPM, 8.65D = (MPH X 38.86)
" " " 12.75D = (MPH X 26.365)
" " " 20.00D = (MPH X 16.81)

NOTE: 1) DATA CORRECTED FOR 1/2 OF ZERO SHIFT.
2) LIKE LETTERS DENOTE APPROX. SAME MC LOAD INDEX.

TEST IDENTIFICATION

TEST #	DATE	AMB. TEMP (°F)	MOTORCYCLE	NO. OF ROLLS	ROLL DIA. (IN)	ROLL SPACE ℄ TO ℄ (IN)	TIRE PRESS (PSIG)	FAU SPEED ROLL (RPM)	FAU TORQUE (#FT)	FAU POWER (HP)	MC SPEED (RPM)	MC TORQUE (#FT)	MC POWER (HP)	Δ TORQUE MC-FAU (#FT)	Δ POWER MC-FAU (HP)	LOAD INDEX
✓ 5	15 MAR 75	66	HONDA 90	2	12.75	17.69	20	256	—	—	148	9.00	0.25	—	—	-
								794	—	—	461	11.28	0.99	—	—	-
								1081	10.35	2.13	636	28.40	3.44	18.05	1.31	B
								778	24.30	3.60	467	50.20	4.46	25.90	0.86	C
6	15 MAR 75	66	HONDA 90	2	12.75	17.69	14	1065	10.80	2.19	633	29.80	3.59	19.00	1.40	B
								770	24.50	3.59	467	52.20	4.64	27.70	1.05	C
								256	—	—	151	9.20	0.26	—	—	-
								812	—	—	477	10.20	0.93	—	—	-
7	15 MAR 75	68	HONDA 90	1	12.75	—	14	243	—	—	144	9.28	0.25	—	—	-
								736	—	—	444	9.08	0.77	—	—	-
								1036	11.60	2.29	628	27.80	3.32	16.20	1.03	B
								807	24.25	3.73	493	48.40	4.54	24.15	0.81	C
8	15 MAR 75	70	HONDA 90	1	12.75	—	20	1115	12.50	2.65	667	30.00	3.81	17.50	1.16	B
								796	24.90	3.77	481	47.60	4.36	22.70	0.59	C
								664	—	—	396	8.92	0.67	—	—	-
								279	—	—	168	8.12	0.26	—	—	-
9	15 MAR 75	70	HONDA 90	1	20.00	—	20	491	—	—	459	7.48	0.65	—	—	-
								165	—	—	153	6.40	0.19	—	—	-
								661	19.05	2.40	620	25.52	3.01	6.47	0.61	B
								498	41.10	3.90	472	46.60	4.19	5.50	0.29	C
10	17 MAR 75	60	HONDA 90	1	20.00	—	20	649	23.90	2.95	615	29.92	3.50	6.02	0.55	B
								494	41.35	3.89	473	47.20	4.25	5.85	0.36	C
								180	—	—	169	6.72	0.22	—	—	-
								410	—	—	387	6.92	0.51	—	—	-
11	17 MAR 75	63	HONDA 90	1	20.00	—	14	188	—	—	181	7.68	0.26	—	—	-
								408	—	—	391	8.72	0.65	—	—	-
								783	—	—	740	8.12	1.14	—	—	-
								664	20.45	2.59	635	28.08	3.40	7.63	0.61	B
								461	10.60	0.93	411	17.06	1.34	6.46	0.41	B
								493	40.90	3.94	476	46.88	4.25	5.98	0.31	C

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MOTORCYCLE TEST SUMMARY

NOTE: 1) DATA CORRECTED FOR ½ OF ZERO SHIFT.
2) LIKE LETTERS DENOTE APPROX. SAME MC LOAD INDEX.

CONSTANTS:
HONDA WHEEL RPM = (COUNT X 1.3333)
HARLEY D. WHEEL RPM = (COUNT X 1.1765)

FAU ROLL RPM, 8.65D = (MPH X 38.[illegible])
" " " 12.75D = (MPH X 26.365)
" " " 20.00D = (MPH X 16.81)

27 MAR 75 PHW
NO # 323
SHEET 3 OF 5

TEST IDENTIFICATION

TEST #	DATE	AMB. TEMP (°F)	MOTORCYCLE	NO. OF ROLLS	ROLL DIA. (IN)	ROLL SPACE ℄ TO ℄ (IN)	TIRE PRES. (PSIG)	FAU SPEED ROLL (RPM)	FAU TORQUE (#FT)	FAU POWER (HP)	MC SPEED (RPM)	MC TORQUE (#FT)	MC POWER (HP)	Δ TORQUE MC-FAU (#FT)	Δ POWER MC-FAU (HP)	LOAD INDEX
12	18MAR75	74	HARLEY DAVIDSON FLH	1	20.00	–	26	704	34.45	4.62	582	53.72	5.92	19.27	1.30	D
								449	13.55	1.16	372	25.85	1.83	12.30	0.67	D
								190	3.25	0.12	156	13.18	0.39	9.93	0.27	D
								999	118.10	22.46	828	163.70	25.81	45.00	3.35	E
								605	107.25	12.35	507	146.97	14.19	39.[illegible]2	1.84	F
13	20MAR75	68	H.D. FLH	1	20.00	–	26	699	32.30	4.30	585	53.21	5.93	20.91	1.63	D
								429	12.95	1.06	360	27.87	1.91	14.92	0.85	D
								175	2.65	0.09	147	13.18	0.37	10.55	0.28	D
								1009	119.40	22.94	842	168.00	26.93	48.60	3.99	E
								602	41.30	4.73	505	66.14	6.36	24.84	1.63	E
								348	13.05	0.86	291	28.63	1.59	15.58	0.73	E
14	20MAR75	68	H.D. FLH	1	20.00	–	26	699	31.75	4.23	585	54.73	6.10	22.98	1.87	D
								445	12.50	1.06	368	28.63	2.01	16.13	0.95	D
								175	2.20	0.07	144	13.18	0.36	10.98	0.29	D
								1009	107.65	20.68	841	161.16	25.81	53.51	5.13	E
								602	37.45	4.29	502	61.32	5.86	23.87	1.57	E
								326	10.60	0.66	273	24.83	1.29	14.23	0.63	E
15	20MAR75	71	H.D. FLH	1	20.00	–	26	708	29.85	4.02	579	57.02	6.29	27.17	2.27	D
								422	13.45	1.08	348	26.45	1.75	13.00	0.67	D
								177	2.80	0.09	145	12.92	0.36	10.12	0.27	D
								1007	130.95	25.11	840	167.09	26.72	36.14	1.61	E
								609	46.70	5.42	506	65.88	6.35	19.18	0.93	E
								439	22.90	1.91	364	37.50	2.60	14.60	0.69	E
16	21MAR75	71	H.D. FLH	1	12.75	–	26	1070	22.05	4.49	573	52.96	5.78	30.91	1.29	D
								770	10.30	1.51	411	31.42	2.46	21.12	0.95	D
								395	3.05	0.23	214	17.48	0.71	14.43	0.48	D
								1595	82.65	25.10	858	167.09	27.30	84.44	2.20	E
								960	29.40	5.37	513	63.60	6.21	34.20	0.84	E
								667	13.30	1.69	355	35.22	2.38	21.92	0.69	E

MOTORCYCLE TEST SUMMARY

NOTE: 1) DATA CORRECTED FOR ½ OF ZERO SHIFT.
2) LIKE LETTERS DENOTE APPROX. SAME MC LOAD INDEX.

CONSTANTS: HONDA WHEEL RPM = (COUNT X 1.3333)
HARLEY D. WHEEL RPM = (COUNT X 1.1765)

FAJ ROLL RPM, 8.65D = (MPH X 38.86)
" " " 12.75D = (MPH X 26.36)
" " " 20.0 D = (MPH X 16.81)

27 MAR 75 FAW
NC # 33[illegible]
SHEET 4 OF 5

TEST IDENTIFICATION

TEST #	DATE	AMB. TEMP (°F)	MOTORCYCLE	NO. OF ROLLS	ROLL DIA. (IN)	ROLL SPACE ℄ TO ℄ (IN)	TIRE PRESS (PSIG)	FAJ [illegible] ROLL (RPM)	FAJ TORQUE (#FT)	FAJ POWER (HP)	MC SPEED (RPM)	MC TORQUE (#FT)	MC POWER (HP)	Δ TORQUE MC-FAJ (#FT)	Δ POWER MC-FAJ (HP)	LOAD INDEX
17	21 MAR 75	65	H.D. FLH	2	12.75	19.56	26	1078	19.15	3.93	568	52.71	5.70	33.56	1.77	D
								664	7.35	0.93	347	26.51	1.75	19.16	0.82	D
								385	4.65	0.34	204	20.02	0.78	15.37	0.44	D
								1614	76.20	23.42	862	167.85	27.55	91.65	4.13	E
								949	25.00	4.52	500	60.82	5.79	35.82	1.27	E
								746	15.10	2.14	393	39.89	2.98	24.79	0.84	E
18	24 MAR 75	71	H.D. FLH	1	8.65	–	26	1582	12.90	3.89	579	51.54	5.68	38.64	1.79	D
								1003	5.30	1.01	367	26.35	1.84	21.05	0.83	D
								630	2.60	0.31	229	19.87	0.87	17.27	0.56	D
								2308	56.30	24.74	853	166.48	27.04	110.18	2.30	E
								1422	20.20	5.47	519	65.88	6.51	45.68	1.04	E
								960	8.70	1.59	346	35.32	2.33	26.62	0.74	E
19	24 MAR 75	74	H.D. FLH	2	8.65	17.56	26	1605	12.30	3.76	573	52.86	5.77	40.56	2.01	D
								1038	5.20	1.03	369	29.39	2.06	24.19	1.03	D
								672	2.80	0.36	238	22.81	1.63	20.01	0.67	D
								2312	49.95	21.99	849	166.33	26.89	116.38	4.90	E
								1411	17.30	4.65	507	64.87	6.26	47.57	1.61	E
								960	7.55	1.38	338	37.10	2.39	29.55	1.01	E
20	24 MAR 75	73	H.D. FLH	2	8.65	17.56	26	1756	–	–	628	20.79	2.49	–	–	–
								1104	–	–	391	14.70	1.09	–	–	–
								614	–	–	215	13.68	0.56	–	–	–
21	24 MAR 75	76	H.D. FLH	2	8.65	11.38	26	1912	–	–	681	17.84	2.31	–	–	–
								1111	–	–	398	14.19	1.08	–	–	–
								672	–	–	236	12.67	0.57	–	–	–
22	25 MAR 75	62	HONDA 90	1	8.65	–	20	1585	7.35	2.22	644	29.4	3.61	22.05	1.39	B
								968	3.4	0.63	392	18.64	1.39	15.24	0.76	B
								443	1.35	0.11	179	14.32	0.49	12.97	0.38	B
								1558	10.65	3.16	640	37.36	4.55	26.71	1.39	G
								1026	5.20	1.02	417	22.96	1.82	17.16	0.80	G
								521	1.85	0.18	213	14.60	0.59	12.75	0.41	G

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28 MAR 75 PHr
WO # 333
SHEET 5 OF 5

MOTORCYCLE TEST SUMMARY

NOTE: 1) DATA CORRECTED FOR ½ OF ZERO SHIFT.
2) LIKE LETTERS DENOTE APPROX. SAME M/C LOAD INDEX.

CONSTANTS: HONDA WHEEL RPM = (COUNT X 1.3333)
HARLEY D. WHEEL RPM = (COUNT X 1.1765)

PAU ROLL RPM, 8.65 D = (MPH X 38.86)
" " " 12.75 D = (MPH X 26.365)
" " " 20.00 D = (MPH X 16.81)

TEST IDENTIFICATION																
TEST #	DATE	AMB. TEMP (°F)	MOTORCYCLE	NO. OF ROLLS	ROLL DIA. (IN)	ROLL SPACE ℄ TO ℄ (IN)	TIRE PRESS (PSIG)	PAU SPEED ROLL (RPM)	PAU TORQUE (#FT)	PAU POWER (HP)	MC SPEED (RPM)	MC TORQUE (#FT)	MC POWER (HP)	Δ TORQUE MC - PAU (#FT)	Δ POWER MC-PAU (HP)	LOAD INDEX
23	25MAR75	65	HONDA 90	2	8.65	15.56	20	1574	7.40	2.22	631	35.96	4.32	28.56	2.10	G
								964	3.45	0.63	385	22.72	1.67	19.27	1.04	G
								462	1.45	0.13	185	17.00	0.60	15.55	0.47	G
								1585	4.60	1.39	633	28.48	3.43	23.38	2.04	B
								1026	1.80	0.35	409	19.60	1.53	17.80	1.18	B
								377	0.50	0.04	148	15.88	0.45	15.38	0.41	3
24	25MAR75	67	HONDA 90	1	20.00	–	20	681	20.60	2.67	639	28.52	3.47	7.92	0.80	B
								457	10.30	0.90	431	16.96	1.39	6.66	0.49	B
								168	2.50	0.08	157	7.96	0.24	5.46	0.16	B
								666	30.00	3.80	627	36.04	4.30	6.04	0.50	G
								495	14.00	1.19	419	19.76	1.58	5.76	0.39	G
								156	2.40	0.07	145	7.84	0.22	5.44	0.15	G